PATENT

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

Application No.:	10/537,566
Filing Date:	November 9, 2005
Applicant:	Christopher P. Revill, et al.
Group Art Unit:	3616
Examiner:	Karen J. Amores
Title:	HYDRAULIC SUSPENSION SYSTEM
Attorney Docket:	1316K-000028/NP

Commissioner for Patents
P.O. Box 1450
Alexandria, Virginia 22313-1450

RESPONSE

Sir:

In response to the Notification of Non-Compliant Appeal Brief, enclosed is an amended Brief which identifies the claims that are being appealed in the "Status of Claims" section of the Brief. Applicants believe the Appeal Brief is now in compliance with 37 CFR 41.37(c).

If the Examiner believes that personal communication will expedite prosecution of this application, the Examiner is invited to telephone the undersigned at (248) 641-1600.

Respectfully submitted,

Dated: April 14, 2009

By: ______________________
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PATENT

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE
BEFORE THE BOARD OF PATENT APPEALS AND INTERFERENCES

Group Art Unit:	3616	)	
Examiner:	Karen J. Amores	)	APPEAL BRIEF
Appellants:	Christopher P. Revill, et al.	)	Appeal No.
Serial No.:	10/537,566	)	
Filed:	November 9, 2005	)	
For:	HYDRAULIC SUSPENSION SYSTEM	)	
Attorney Docket:	1316K-000028/NP	)	

Michael J. Schmidt

For Appellants

APPELLANTS' APPEAL BRIEF

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Dear Sir:

This is an Appeal of the rejection of Claims 1, 4, 6, 8-12 and 17 under 35 U.S.C. §103(a) as being unpatentable over Heyring, et al., U.S. 6,270,098 ("Heyring '098") in view of Heyring, et al., U.S. 6,761,371 ("Heyring '371") and the rejection of Claim 18 under 35 U.S.C. §103(a) as being unpatentable over Heyring '098 and Heyring '371 as applied to Claims 6 and 17 above, and further in view of Kobayashi, U.S. 7,210,688 ("Kobayashi").

REAL PARTY OF INTEREST

Kinetic Pty Ltd is the real party of interest in the present application. Kinetic Pty Ltd is the Assignee of the present application as recorded with the United States Patent and Trademark Office on November 17, 2005 on Reel 016792, Frame 0319.

RELATED APPEALS AND INTERFERENCES

To the best of Appellants' knowledge, no other appeals or interferences are pending which will directly affect, be directly affected by or have a bearing on the Boards decision in the present pending appeal.

STATUS OF THE CLAIMS

Claims 1, 4, 6, 8-12 and 17 are rejected under 35 U.S.C. § 103(a) as being unpatentable over Heyring, et al. U.S. 6,270,098 ("Heyring '098") in view of Heyring, et al. U.S. 6,761,371 ("Heyring '371").

Claim 18 is rejected under 35 U.S.C. § 103(a) as being unpatentable over Heyring '098 and '371 as applied to Claims 6 and 17 above, and further in view of Kobayashi, U.S. 7,210,688 ("Kobayashi").

Claims 2, 3, 5, 13-16, 19-22 and 24-33 are objected to as being dependent upon a rejected base claim, but would be allowable if rewritten in independent form including all of the limitations of the base claim and any intervening claims.

Claims 7 and 34-36 are allowed.

Applicants are appealing the rejections of Claims 1, 4, 6, 8-12, 17 and 18.

STATUS OF THE AMENDMENTS

A Final Office Action was mailed on June 5, 2008.

Appellants filed a response to the Final Office Action on September 4, 2008 amending the claims to correct an issue relating to antecedent basis.

An Advisory Action was mailed on September 15, 2008 stating that our response was entered but it was not considered as placing the application in condition for allowance.

On October 6, 2008, Appellants filed a Notice of Appeal and Pre-Appeal Brief Review Request.

A Notice of Panel Decision from the Pre-Appeal Brief Review was mailed October 27, 2008 stating that there is at least one actual issue for appeal and the appeal would proceed to the Board of Patent Appeal and Interferences.

SUMMARY OF THE CLAIMED INVENTION

The summary of the claimed invention will refer to U.S. Publication 2006/0151969 which is the U.S. publication of PCT/AU03/01637. The pending U.S. application is a 371 of PCT/AU03/01637.

Referring now to Figure 1 and to paragraphs [0060], [0061], [0062] and [0067], Claim 1 defines a vehicle suspension system having a damping and stiffness system. The vehicle includes a first pair of diagonally spaced wheel assemblies 11, 13 and a second pair of diagonally spaced wheel assemblies 12, 14. The vehicle suspension also includes front and rear resilient support means 27, 28, 29 and 30 for supporting the vehicle above the wheel assemblies.

The damping and stiffness system includes at least one wheel ram 11, 12, 13 and 14 between each wheel assembly and the vehicle body and each ram includes at least one compression chamber 45, 46, 47 and 48. A load distribution unit 76 is interconnected between the compression chambers of the wheel rams. The load distribution unit including two piston rod assemblies 97 and 98, first, second, third and fourth system volumes 89, 92, 90 and 91 and first and second modal resistance volumes 93, 94 and 95, 96.

The first piston rod assembly 97 defines first, second, third and fourth effective areas and the second piston rod assembly 98 defines fifth, sixth, seventh and eighth effective areas. The first effective area defines a movable wall of the first system volume 89, the second effective area defines a movable wall of the second system volume 92, the third effective area defines a movable wall of the first modal resistance volume 93 and the fourth effective area defines

a movable wall of the second modal resistance volume 96. The fifth effective area defines a movable wall of the third system volume 90, the sixth effective area defines a movable wall of the fourth system volume 91, the seventh effective area defines a movable wall of the first modal resistance volume 94 and the eighth effective area defines a movable wall of the second modal resistance volume 95.

The first system volume 89 increases in volume proportionately to the decrease in volume of the second system volume 92 and the third system volume 90 increases in volume proportionately to the decrease in volume of the fourth system volume 91. The volume of the first modal resilience volume 93, 94 decreasing in volume proportionately to the increase in volume of the first and third system volumes 89, 90 and the volume of the second modal resilience volume decreasing proportionately to the increase in volume of the second and fourth system volumes 91, 92.

The first and fourth system volumes 89, 91 are connected to the compression chambers of one of the pairs of diagonally spaced wheel assemblies 11, 13 and the second and third system volumes are connected to the compression chambers of the other pair of diagonally spaced wheel assemblies 12, 14. The damping system thereby providing substantially zero warp stiffness.

The vehicle is primarily supported by the vehicle resilient support means 27, 28, 29 and 30 which is functionally separate from the damping and stiffness system.

Referring now to Figure 1 and to paragraphs [0060], [0061], [0062] and [0067], Claim 6 defines a vehicle suspension system having a damping and stiffness system. The vehicle includes at least two forward wheel assemblies 11, 12 and at least two rearward wheel assemblies 13, 14. The vehicle suspension system includes front and rear resilient support means 27, 28, 29 and 30 for supporting the vehicle above the wheel assemblies.

The damping and stiffness system includes at least two front 11, 12 and two rear 13, 14 wheel rams located between the wheel assemblies and the vehicle body and each ram including at least a compression chamber 45, 46, 47 and 48. A load distribution unit 76 includes a first pair of axially aligned primary chambers 89, 92 and a second pair of axially aligned primary chambers 90, 91. Each primary chamber including a piston 97, 98 separating each primary chamber into secondary chambers 89, 93; 92, 96; 90, 94; and 91, 95.

One of the secondary chambers 89 in the first pair of primary chambers being a first front system chamber connected to the compression chamber 45 of a front wheel ram on a first side of the vehicle. The other secondary chamber 96 in the first pair of primary chambers being a first back pitch chamber.

The other of the secondary chambers 92 in the first pair of primary chambers being a first back system chamber and being connected to a back wheel ram 18 on the first side of the vehicle. The other secondary chamber 93 in the first pair of primary chambers being a first front pitch chamber.

One of the secondary chambers 90 in the second pair of primary chambers being a second front system chamber connected to the compression

chamber 46 of a front wheel ram on a second side of the vehicle. The other secondary chamber 95 in the second pair of primary chambers being a second back pitch chamber.

The other of the secondary chambers 91 in the second pair of primary chambers being a second back system chamber and being connected to a back wheel ram 17 on the second side of the vehicle. The other secondary chamber 94 in the second pair of primary chambers being a first front pitch chamber.

The first and second front pitch chambers 93, 94 being interconnected to form a front pitch volume and the first and second back pitch chambers 95, 96 being interconnected to form a back pitch volume.

The vehicle is primarily supported by the vehicle resilient support means which is functionally separate from the damping and stiffness system.

GROUNDS OF REJECTION TO BE REVIEWED ON APPEAL

Appellants present the following issues for review:

1) Claims 1, 4, 6, 8-12 and 17 are rejected under 35 U.S.C. § 103(a) as being unpatentable over Heyring, et al. U.S. 6,270,098 ("Heyring '098") in view of Heyring, et al. U.S. 6,761,371 ("Heyring '371"). Claim 18 is rejected under 35 U.S.C. § 103(a) as being unpatentable over Heyring '098 and '371 as applied to Claims 6 and 17 above, and further in view of Kobayashi, U.S. 7,210,688 ("Kobayashi").

ARGUMENT

Claims 1, 4, 6, 8-12, 17 and 18

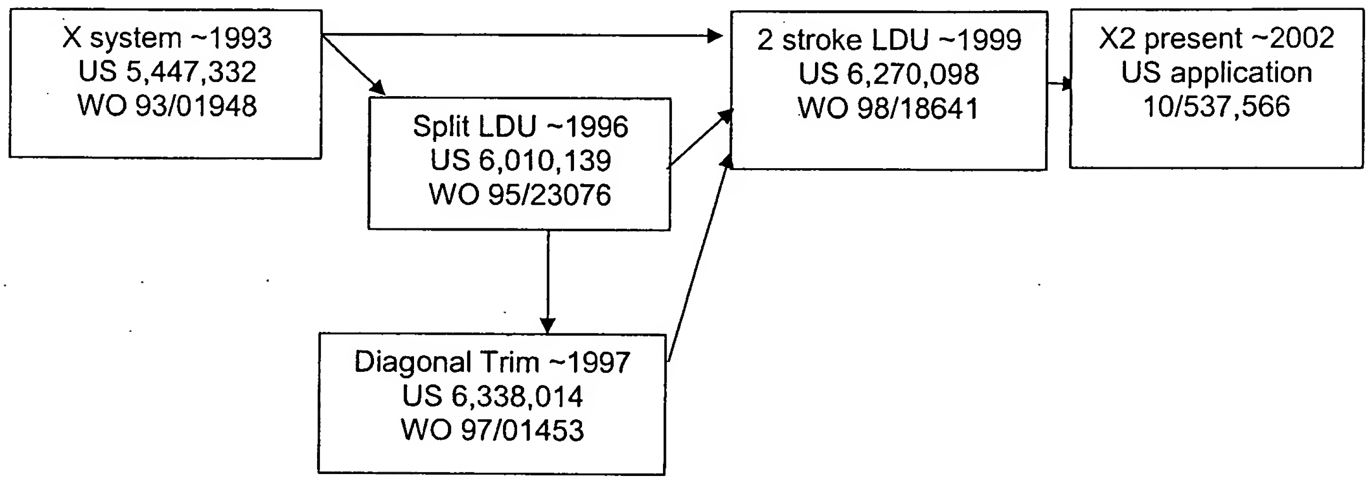
Claims 1, 4, 6, 8-12 and 17 are rejected under 35 U.S.C. § 103(a) as being unpatentable over Heyring, et al. U.S. 6,270,098 ("Heyring '098") in view of Heyring, et al. U.S. 6,761,371 ("Heyring '371"). Claim 18 is rejected under 35 U.S.C. § 103(a) as being unpatentable over Heyring '098 and '371 as applied to Claims 6 and 17 above, and further in view of Kobayashi, U.S. 7,210,688 ("Kobayashi").

Independent Claims 1 and 6 each define that the vehicle is primarily supported by the vehicle resilient support means (27, 28, 29 and 30 of Figure 1) and that the vehicle resilient support means is functionally separate from the damping and stiffness system. Thus, the damping and stiffness system of the present invention provides little or no support for the vehicle, the vehicle resilient support means provides this support function. The primary function of the damping and stiffness system is to provide damping and stiffness for the vehicle's suspension system.

The Examiner agrees that Heyring '098 does not directly disclose a front and rear vehicle resilient means. The Examiner then looks to Heyring '371 which teaches a vehicle suspension system including front and rear vehicle resilient support means (17) between the vehicle body and the wheel assemblies for resiliently supporting the vehicle above the wheel assemblies, wherein the vehicle is primarily supported by the vehicle resilient support means which is functionally separate from the damping and stiffness system. The Examiner's

position is that it would have been obvious to modify Heyring '098 such that it comprised front and rear vehicle resilient support means in view of Heyring '371 so as to provide independent support means capable of supporting the entire weight of the vehicle (column 11, line 7). Appellants respectfully traverse this position of the Examiner.

The invention in present application evolved through a technology family developed by Kinetic Pty Ltd, the development timeline and relationship is as follows:



Systems Supporting the Vehicle

Working backwards through this history and cases;

U.S. 6,270,098

The cited '098 (2 stroke LDU) incorporates by reference (see column 1, lines 15-26) Kinetic's international application number PCT/AU95/00096 (WO 95/23076 – split LDU) – which eventually became U.S. 6,010,139. The invention described and defined in '098 is set out and described as an improved

arrangement of the load distribution unit of PCT/AU95/00096 (see column 1, lines 29-34) (WO 95/23076 and U.S. 6,010,139).

U.S. 6,010,139

U.S. 6,010,139 provides an improved vehicle suspension system with more optimal relationship between the pitch and the roll control of the vehicle (column 3, lines 20-23). This is achieved through an improved load distribution unit (LDU) in the light of and in some great detail over and above the vehicle hydraulic suspension system set out in international patent application no. PCT/AU92/00362 (published as WO 93/01948 – and which also eventually became U.S. 5,447,332 known to Kinetic as their X system - as shown above).

U.S. 6,010,139 clearly explains at column 1, lines 61-65 "This prior proposed vehicle suspension system [of WO 93/01948; U.S. 5,447,332] obviates the use of ordinary springs (e.g. coils, leaf, or torsion bar springs) as well as conventional telescopic dampers (commonly referred to as shock absorbers) and roll or sway stabilizer bars. '139 goes on to state at column 1, line 66 to column 2, line 1 that "Springing or resilience is provided by way of gas filled accumulators..." in the referenced system disclosed in WO 93/01948.

U.S. 5,447,332

U.S. 5,447,332 discloses a hydraulic suspension system which "<u>eliminates the use of conventional spring components and incorporates a totally fluid suspension</u>..." (see column 2, lines 22-29).

Column 3, lines 1-7 further explains "The transference of the fluid medium at balancing pressures between the rams of the diagonally connected front and rear wheels <u>effects leveling and stabilization of the vehicle body by the extension and retraction of the appropriate rams which provide the changing adjustable support mechanism of the vehicle body with reference to the unsprung wheel assemblies</u>".

U.S. 5,447,332 is itself an improvement over a conventional diagonally interconnected vehicle hydraulic suspension system, such as in FR 1535641 published August 9, 1968 to Industrial Development Company Establishments of Liechtenstein. The entire purpose of the invention defined and disclosed in '332 is to provide a load distribution unit (LDU) interposed between two diagonal fluid circuits of such suspension system. This provides a pressure balance between the circuits and removes cross axle articulation (warp) stiffness whilst increasing roll and heave stiffness.

Evolution in Development

Thus,

U.S. 5,447,332 discloses a base chassis supporting hydraulic suspension system without requiring separate support means.

U.S. 6,010,139 provides an improvement to that base system of U.S. 5,447,332 by providing an improved load distribution unit arrangement incorporating additional pitch resilience.

U.S. 6,270,098 provides an improved LDU arrangement over U.S.

6,010,139.

Each of these patents U.S. 6,270,098, U.S. 6,010,139 and U.S. 5,447,332 and the systems/arrangements they disclose relate directly to and for a self supporting hydraulic suspension system not requiring additional resilient supports for the vehicle body.

Thus, the Heyring '098 patent teaches against the addition of the resilient means in Heyring '371 because the system in Heyring '098, which was derived from the 6,010,139 patent which is incorporated by reference in Heyring '098, states that the suspension system obviates the use of ordinary springs. Thus, there is no suggestion to combine Heyring '371 with Heyring '098.

Suspension System

A vehicle suspension system supports a vehicle body and any load. Static and dynamic support for the vehicle body and any load is the essence of what a suspension system does/is.

U.S. 6,270,098 is directed to "A load distribution unit for a vehicle suspension system" (column 1, lines 36-37). That is, a load distribution unit (LDU) applicable to a vehicle hydraulic suspension system. The vehicle hydraulic suspension system supports the vehicle body and any load, and the LDU controls/transfers dynamic forces within the system.

U.S. 6,010,139 is directed to "A suspension system for a vehicle" (column 1, lines 20-24) with features of a hydraulic suspension system. The suspension system is a hydraulic system that supports the vehicle body and any load.

U.S. 5,447,332 is directed to a vehicle suspension system which

eliminates the use of conventional spring components..." (column 2, lines 22-24) i.e. is a hydraulic suspension system.

None of these documents teaches or discloses a vehicle hydraulic damping and stiffness system and a functionally separate resilient vehicle support means (coil or leaf springs etc).

The present application 10/537,566 discloses an invention which is an improvement over and above these prior systems and arrangements and is not relevant to such chassis supporting suspension systems i.e. to systems where separate support means for the vehicle body are not provided or disclosed.

The appellants do not believe that the addition of separate support means to Heyring '098 brings results that would have been immediately apparent and predictable to the relevant skilled person at the December 6, 2002 priority date of the present application.

It is worth remembering that Heyring '098 is owned by Kinetic Ltd (now Kinetic Pty Ltd - "Kinetic"). The invention defined in the present application is an improvement over that Heyring '098 technology. Kinetic is the leading proponent in this field of interconnected hydraulic suspension systems and damping systems, and developed the present invention as a technically different system to that of '098, with associated different benefits and advantages.

6,270,098 System

The Examiner maintains that Heyring '098 could be provided with separate resilient support means from Heyring '371 (also assigned to Kinetic) to meet the features of present claim 1 and many dependent claims.

- The prospect of a system according to Heyring '098 having separate resilient support means is irrational given that the system of Heyring '098 itself supports the vehicle body without separate resilient support means (the system hydraulic rams supporting the body – as previously argued) – The hydraulic system of Heyring '098 provides full support for the vehicle chassis/body, as demonstrated in that document by
    - The rams 1-4 in Figure 1 having broad piston rods, and the relative difference in effective piston in the compression and rebound chambers thereof
    - Importantly, reference at column 1, lines 24-25 of Heyring '098 incorporates by reference PCT/AU95/00096 (published as WO 95/23076 and US 6,010,139). This reference is directed to a hydraulic suspension system with a load distribution unit. The suspension system is arranged to support the vehicle body, as clearly disclosed at page 6 lines 13-15 of the PCT document WO 95/23076 *"the hydraulic rams may be of either double or single acting type.* ***In either arrangement, the chambers of the rams that are providing the support for the vehicle*** *are connected to the balance means."* Heyring '098 seeks to provide an improved load distribution unit for such a system, as per column 1, lines 29-34 of Heyring '098.

Problems with 6,270,098 type systems

- o temperature changes cause volume changes in the hydraulic fluid, leading to changes in vehicle body ride height. Consequently, a control system is required to correct for excessive fluid volume changes.
- o the control system must regulate primarily on the basis of vehicle height (as per US Patent No. 6,338,014 to Heyring et al) as the pressure in each fluid volume is dependent on the load and load distribution on the vehicle body.
- o regulating vehicle height with changing load means that when sizing the components of the hydraulic system, the load variation and wheel travel dictate the cylinder bore and rod diameters and accumulator sizes, in addition to the requirements of heave stiffness, roll stiffness, etc. If the load variation is too large relative to the static load supported by the hydraulic system, the limit of feasible system parameters is exceeded.

Heyring '098 would not be modified with additional supports

- As mentioned above, the system of Heyring '098 itself already supports the vehicle body. The addition of further supports in the form of resilient support means, such as coil or leaf springs, adding unnecessary weight and cost to the vehicle, with compromise in functionality, would not be relevant or sensible.

- If additional support means are provided to the hydraulic system of Heyring '098 such that the additional support means provide the primary support of the vehicle body and the hydraulic system thereby no longer provides a significant portion of the support of the vehicle body, it:
    - cannot be used for load levelling because, if there is negligible load initially supported by the hydraulic system, any additional load supported by the system would be a significant proportion of, or indeed many times the initial load. It is not possible to design a hydraulic system using a single fixed set of resilient means such as accumulators that can provide suitable stiffness over such an extreme load range.
    - cannot provide significant heave stiffness because in providing little or negligible support of the vehicle body, the pressure in the hydraulic system and/or the effective rod area of each wheel ram must be significantly lower than that of a supporting system such as Heyring '098 (so that the push-out force is low permitting the separate support means to provide the support of the vehicle body as per claim 1). In that case there is little change in push-out force with heave motions of the vehicle body with respect to the wheels, ie very low heave stiffness. For reference the operating pressure is typically 3-5 times lower and the effective rod area is typically 2.5-5 times smaller.

- must be designed differently to produce the different required functionality (coil springs cannot be added to an existing '098 function system) as a hydraulic system which provides very low or negligible push-out force and heave stiffness whilst providing significant roll (and pitch) stiffness (as in the present invention) by definition has a high ratio of roll stiffness to heave stiffness.
    - As noted above, the Heyring '098 system requires larger effective rod diameters to provide the support function, so the ratio of roll stiffness to heave stiffness required is relatively low (typically less than 2.5:1). In the present invention a minimal push-out force is required, so the ratio of roll stiffness to heave stiffness is extremely high (typically over 7.5:1).
    - The ratio of roll stiffness to heave stiffness is determined by a function of rod and bore areas i.e. ratio of (over piston area + under piston area) : rod area. That is, it is determined by the components.
    - Thus, attempting to use the components from the Heyring '098 system in the hydraulic system of the present invention would not achieve the aim of high roll stiffness with minimal heave stiffness without significant redesign to completely change the ratio of roll stiffness to heave stiffness. The

components therefore are not intercompatible between Heyring '098 and the present invention.

- must include control based primarily on pressure control rather than primarily on displacement control because in Heyring '098 where the hydraulic system provides the support of the vehicle body, the push out force required by the wheel rams is determined by the mass of the vehicle body and any associated load. Therefore the pressure in the hydraulic system volumes is directly related to and determined by the vehicle body mass and as the system fluid volumes change due to temperature, leakage past seals, etc. the pressure in the system volumes remains the same but the vehicle height changes. Therefore, a displacement control is required to maintain the vehicle in an operating condition. Conversely in the present invention, the hydraulic system does not provide the primary support to the vehicle body so the pressure in the fluid volumes is not inherently determined by the vehicle body mass. Furthermore as there is little push-out force and little heave stiffness, adjusting the system volume pressures has little effect on the vehicle body position. In that case to maintain the volume of fluid in each system at an amount to maintain the operation of the hydraulic system and to keep the vehicle in an operating condition, a displacement control of the wheel rams is inappropriate and a pressure control of the system volume pressures is necessary.

Points of differentiation

It is not normal to use a first support means for the vehicle body and a second additional support means. If coil springs are supporting the vehicle, leaf springs or additional coil springs are not required. The support is already there.

Contrary to the Examiner's point of view in the recent Advisory Action, shock absorbers are not resilient supports. Shock absorbers are used in combination with coil or leaf springs as they provide completely separate functions - shock absorbers provide damping with negligible support or heave stiffness and the coil or leaf springs provide support and heave stiffness with negligible damping.

As the hydraulic system of Heyring '098 provides all of the support for the vehicle body, not only is there no motivation to provide additional support means, but as discussed above, the hydraulic system would not operate if simply coil springs or leaf springs were added. The hydraulic system would need to be completely redesigned to reduce its functionality (ie remove virtually all of the heave stiffness and supporting push out force). This is supported by the final paragraph of claim 1 in which <u>the vehicle resilient support means is functionally separate from the damping and stiffness system</u>.

This reduced functionality of the hydraulic system is fundamental to the new features and benefits provided by the present invention over Heyring '098. Unless Heyring '098 is redesigned to remove virtually all of the push out force, the ratio between roll and heave stiffness is insufficient to enable the hydraulic system to provide sufficient roll stiffness without providing a significant push out

force and heave stiffness.

Thus, the key novel and inventive feature of the hydraulic system of the present invention is that "...the vehicle is primarily supported by the vehicle resilient support means which is functionally separate from the damping and stiffness system" i.e. the hydraulic systems provides minimal / negligible supporting push-out force or heave stiffness, which in turn brings a range of benefits not possible with Heyring '098:

- powered fluid pressure supply system not required to maintain vehicle at ride height as temperature changes, etc.
- a very high ratio of heave to roll (and pitch) stiffness meaning that roll and pitch stiffness rates can be adjusted independent of load (in Heyring '098 they vary with load but are fixed for any given load), or the control system can be simplified to remove a significant portion of system cost.
- the hydraulic system components can have a lower pressure rating, lighter weight and reduced cost compared to those of Heyring '098.

The system disclosed in Heyring '098 completely decouples warp (cross axle articulation) from every other suspension mode (heave, pitch, roll etc). This is achieved by not having separate support means, and also saves on cost and weight of separate support means. There is no motivation in Heyring '098 or 6,010,139 to add separate resilient support means that would otherwise remove this freedom of cross axle articulation (i.e. would add warp stiffness). The system of Heyring '098 tries to remove warp stiffness not add it in, and it would

therefore be counterintuitive for the skilled addressee to want to add additional separate support means to a system that:

1. already is chassis supporting, and
2. tries to remove warp stiffness.

In addition, the system of Heyring '098 requires high pressure to support the weight of the vehicle body. This leads to high cost, high specification components that need to be able to cope with those high pressures.

Also, "stiction" becomes an issue in the Heyring '098 system. Stationary friction ("stiction") within the LDU and rams arise because of the high pressures and need for large seals to cope with those pressures. Each system volume has a pressure determined by the load and load position. Therefore each system pressure can be different, resulting in stiction at the largest number of points in the system. Stiction is felt as an undesirable effect on the ride quality of the suspension system.

The high pressures can result in leakage from and within the Heyring '098 system (in between the system volumes due to their differing pressures). This requires additional cost of pressure monitoring and pressure compensation components, as well as high pressure supply equipment. All of this leads to additional cost.

In contrast, the invention defined in the present application provides <u>the vehicle resilient support means functionally separate from the damping and stiffness system</u>. This avoids the complexity, cost and pressure related problems (stiction, leakage etc) of the chassis supporting Heyring '098 system.

Summary

Heyring '098 teaches an improved load distribution device for a hydraulic suspension system. The system being of the type arranged to support the vehicle body (per WO 95/23076 incorporated into Heyring '098).

There is no motivation for the relevant skilled person to provide additional primary resilient supports for the vehicle body when the hydraulic suspension system of Heyring '098 already itself supports the body and teaches against supporting means. Heyring '098 does not teach or suggest the use of additional resilient support means to support the vehicle body.

Whilst Heyring '371 discloses separate resilient support means, there is no motivation within Heyring '098 and Heyring '371 to combine those documents given the differing types of system.

If necessary is a Declaration from one of the inventors, Christopher Revill, which supports the above discussion and explanation, can be filed.

CONCLUSION

Appellants respectfully request the rejections of the Examiner be withdrawn and the allowance of the pending claims.

Respectfully submitted,

Dated: April 14, 2009

By: ____________________
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APPENDIX A

PENDING CLAIMS

1. A vehicle suspension system having a damping and stiffness system for a vehicle, the vehicle including a vehicle body and a first pair and a second pair of diagonally spaced wheel assemblies, the first pair of diagonally spaced wheel assemblies including at least one front left wheel assembly and at least one back right wheel assembly, the second pair of diagonally spaced wheel assemblies including at least one front right wheel assembly and at least one back left wheel assembly, the vehicle suspension system also including front and rear vehicle resilient support means between the vehicle body and the wheel assemblies for resiliently supporting the vehicle above the wheel assemblies, the damping and stiffness system including:

at least one wheel ram located between each wheel assembly and the vehicle body, each ram including at least a compression chamber;

a load distribution unit interconnected between the compression chambers of the front left, front right, back left and back right wheel rams, the load distribution unit including first and second piston rod assemblies, first, second, third and fourth system volumes and first and second modal resilience volumes,

the first piston rod assembly defining first, second, third and fourth effective areas, the second piston rod assembly defining fifth, sixth, seventh and eighth effective areas, the first and second piston rod assemblies being located within the load distribution unit such that each piston rod assembly can rotate

about and slide along a major axis of the piston rod assembly,

the first effective area defines a movable wall of the first system volume such that as the first piston rod assembly slides along its major axis, the volume of the first system volume varies, the second effective area defines a moveable wall of the second system volume, the third effective area defines a movable wall of the first modal resilience volume, the fourth effective area defines a movable wall of the second modal resilience volume, the fifth effective area defines a movable wall of the third system volume such that as the second piston rod assembly slides along its major axis, the volume of the third system volume varies, the sixth effective area defines a moveable wall of the fourth system volume, the seventh effective area defines a movable wall of the first modal resilience volume, and the eighth effective area defines a movable wall of the second modal resilience volume,

the first system volume increasing in volume proportionately to the decrease in volume of the second system volume with motion of the first piston rod assembly, the third system volume increasing in volume proportionately to the decrease in volume of the fourth system volume with motion of the second piston rod assembly,

the volume of the first modal resilience volume decreasing proportionately to the increase in volume of the first and third system volumes with motion of the first and second piston rod assemblies, the volume of the second modal resilience volume decreasing proportionately to the increase in volume of the second and fourth system volumes,

the first and fourth system volumes being connected to the compression chambers of the wheel rams associated with one of the pairs of diagonally spaced wheel assemblies, the second and third system volumes being connected to the compression chambers of the wheel rams associated with the other pair of diagonally spaced wheel assemblies, the damping and stiffness system thereby providing substantially zero warp stiffness; and

wherein the vehicle is primarily supported by the vehicle resilient support means which is functionally separate from the damping and stiffness system.

2. A vehicle suspension system having a damping and stiffness system according to claim 1 further including a pressure maintenance device connected in fluid communication to the first, second, third and fourth system volumes to maintain the static pressure of said system volumes at a substantially common pressure.

3. A vehicle suspension system having a damping and stiffness system as claimed in claim 2 wherein the pressure maintenance device is further connected in fluid communication to the first and second modal resilience volumes to maintain the static pressure of the modal resilience volumes at substantially the same common pressure.

4. A vehicle suspension system having a damping and stiffness system as claimed in claim 1 wherein the first system volume is connected to the compression chamber of the at least one wheel ram associated with the at least one front left wheel assembly, the second system volume is connected to the compression chamber of the at least one wheel ram associated with the at least one back left wheel assembly, the third system volume is connected to the compression chamber of the at least one wheel ram associated with the at least one front right wheel assembly and the fourth system volumes is connected to the compression chamber of the at least one wheel ram associated with the at least one back right wheel assembly,

the first modal resilience volume thereby being a front bump resilience volume and the second modal resilience volume thereby being a back bump resilience volume, the front and back bump resilience volumes thereby providing the damping and stiffness system with additional pitch resilience, independent of the roll and heave stiffness of the damping and stiffness system.

5. A vehicle suspension system having a damping and stiffness system as claimed in claim 1 wherein the first system volume is connected to the compression chamber of the at least one wheel ram associated with the at least one front left wheel assembly, the second system volume is connected to the compression chamber of the at least one wheel ram associated with the at least one front right wheel assembly, the third system volume is connected to the

compression chamber of the at least one wheel ram associated with the at least one back left wheel assembly and the fourth system volume is connected to the compression chamber of the at least one wheel ram associated with the at least one back right wheel assembly,

the first modal resilience volume thereby being a left roll resilience volume and the second modal resilience volume thereby being a right roll resilience volume; the left and right roll resilience volumes thereby providing the damping and stiffness system with additional roll resilience, independent of the pitch and heave stiffness of the damping and stiffness system.

6. A vehicle suspension system having a damping and stiffness system for a vehicle, the vehicle including a vehicle body and at least two forward and two rearward wheel assemblies, the vehicle suspension system also including front and rear vehicle resilient support means between the vehicle body and the wheel assemblies for resiliently supporting the vehicle above the wheel assemblies, the damping and stiffness system including:

at least two front and two rear wheel rams located between the wheel assemblies and the vehicle body, each ram including at least a compression chamber;

a load distribution unit, includes a first pair of axially aligned primary chambers and a second pair of axially aligned primary chambers, each primary chamber including a piston separating each primary chamber into two secondary chambers, a first rod connecting the pistons of the two first primary chambers,

forming a first piston rod assembly and a second rod connecting the pistons of the two second primary chambers forming a second piston rod assembly,

one of the secondary chambers in the first pair of primary chambers being a first front system chamber and being connected to the compression chamber of a front wheel ram on a first side of the vehicle,

the other secondary chamber in the first pair of primary chambers which varies in volume in the same direction as the first front system chamber with motion of the first piston rod assembly, being a first back pitch chamber,

one of the secondary chambers in the first pair of primary chambers which varies in volume in the opposite direction as the first front system chamber with motion of the first piston rod assembly being a first back system chamber and being connected to the compression chamber of a back wheel ram on a first side of the vehicle,

the other secondary chamber in the first pair of primary chambers which varies in volume in the same direction as the first back system chamber with motion of the first piston rod assembly, being a first front pitch chamber,

one of the secondary chambers in the second pair of primary chambers being a second front system chamber and being connected to the compression chamber of a front wheel ram on a second side of the vehicle,

the other secondary chamber in the second pair of primary chambers which varies in volume in the same direction as the second front system chamber with motion of the second piston rod assembly, being a second

back pitch chamber,

one of the secondary chambers in the second pair of primary chambers which varies in volume in the opposite direction as the second front system chamber with motion of the second piston rod assembly being a second back system chamber and being connected to the compression chamber of a back wheel ram on a second side of the vehicle,

the other secondary chamber in the second pair of primary chambers which varies in volume in the same direction as the second back system chamber with motion of the second piston rod assembly, being a second front pitch chamber, and

the first and second front pitch chambers being interconnected forming a front pitch volume and the first and second back pitch chambers being interconnected forming a back pitch volume;

wherein the vehicle is primarily supported by the vehicle resilient support means which is functionally separate from the damping and stiffness system.

7. A vehicle suspension system having a damping and stiffness system for a vehicle, the vehicle including a vehicle body and at least two forward and two rearward wheel assemblies, the vehicle suspension system also including front and rear vehicle resilient support means between the vehicle body and the wheel assemblies for resiliently supporting the vehicle above the wheel assemblies, the damping and stiffness system including:

a load distribution unit, including a first pair of axially aligned primary chambers and a second pair of axially aligned primary chambers, each primary chamber including a piston separating each primary chamber into two secondary chambers, a first rod connecting the pistons of the two first primary chambers, forming a first piston rod assembly and a second rod connecting the pistons of the two second primary chambers forming a second piston rod assembly,

one of the secondary chambers in the first pair of primary chambers being a front left system chamber and being connected to the compression chamber of a front wheel ram on a left side of the vehicle,

the other secondary chamber in the first pair of primary chambers which varies in volume in the same direction as the front system chamber with motion of the first piston rod assembly, being a first right roll chamber,

one of the secondary chambers in the first pair of primary chambers which varies in volume in the opposite direction to the front left system chamber with motion of the first piston rod assembly being a front right system chamber and being connected to the compression chamber of the other front wheel ram on a right side of the vehicle,

the other secondary chamber in the first pair of primary chambers which varies in volume in the same direction as the front right system chamber with motion of the first piston rod assembly, being a first left roll chamber,

one of the secondary chambers in the second pair of primary chambers being a back left system chamber and being connected to the

compression chamber of a back wheel ram on the left side of the vehicle,

the other secondary chamber in the second pair of primary chambers which varies in volume in the same direction as the back left system chamber with motion of the second piston rod assembly, being a second right roll chamber,

one of the secondary chambers in the second pair of primary chambers which varies in volume in the opposite direction as the second front system chamber with motion of the second piston rod assembly being a back right system chamber and being connected to the compression chamber of a back wheel ram on the right side of the vehicle,

the other secondary chamber in the second pair of primary chambers which varies in volume in the same direction as the back right system chamber with motion of the second piston rod assembly, being a second left roll chamber, and

the first and second left roll chambers being interconnected forming a left roll volume and the first and second right roll chambers being interconnected forming a right roll volume;

wherein the vehicle is primarily supported by the vehicle resilient support means which is functionally separate from the damping and stiffness system.

8. A vehicle suspension system having a damping and stiffness system as claimed in claim 6 wherein the wheel rams of at least the two front or

the two rear wheel rams are single-acting rams.

9. A vehicle suspension system having a damping and stiffness system as claimed in claim 8 wherein each single-acting wheel ram includes a piston dividing the ram into a compression and a rebound chamber, damping being provided in the piston of the ram to provide at least a rebound damping force.

10. A vehicle suspension system having a damping and stiffness system according to claim 8 wherein the wheel rams at one end of the vehicle are double-acting wheel rams further including a rebound chamber, the rebound chamber of each double-acting wheel ram being connected to the compression chamber of the diagonally opposite wheel ram.

11. A vehicle suspension system having a damping and stiffness system according to claim 6 wherein each wheel ram is a double-acting ram further including a rebound chamber, the rebound chamber of each double-acting wheel ram being connected to the compression chamber of the diagonally opposite wheel ram.

12. A vehicle suspension system having a damping and stiffness system according to claim 1 wherein the compression chamber of each of at least two of said wheel rams may be in fluid communication with a respective

accumulator.

13. A vehicle suspension system having a damping and stiffness system as claimed in claim 6 wherein the front pitch volume is connected to the back pitch volume through a pitch valve arrangement.

14. A vehicle suspension system having a damping and stiffness system according to claim 13 wherein the pitch valve arrangement includes at least one pitch damper valve to provide pitch damping.

15. A vehicle suspension system having a damping and stiffness system according to claim 14 wherein the at least one pitch damper valve is a variable damper valve.

16. A vehicle suspension system having a damping and stiffness system according to claim 14 wherein the pitch valve arrangement further includes a bypass passage and a bypass valve, the bypass passage being connected to either side of the at least one pitch damper valve, the bypass valve being located in the bypass passage and being switchable to enable or disable the pitch damping.

17. A vehicle suspension system having a damping and stiffness system according to claim 6 wherein the front pitch volume is connected to a front pitch accumulator through a front pitch damper valve and the back pitch volume may be connected to a back pitch accumulator through a back pitch damper valve, the front and back pitch accumulators provide additional pitch resilience in the stiffness and damping system.

18. A vehicle suspension system having a damping and stiffness system according to claim 17 wherein at least one of the front and rear pitch damper valves is a variable damper valve.

19. A vehicle suspension system having a damping and stiffness system according to claim 17 wherein the front pitch volume is connected to the back pitch volume by a pitch stiffness valve.

20. A vehicle suspension system having a damping and stiffness system as claimed in claim 19 wherein the pitch stiffness valve is a damper valve.

21. A vehicle suspension system having a damping and stiffness system as claimed in claim 19 wherein the pitch stiffness valve is a lockout valve to isolate the front pitch volume from the back pitch volume.

22. A vehicle suspension system having a damping and stiffness system according to claim 6 wherein a roll valve is provided to interconnect at least one of the compression chambers of the at least two front wheel rams and the compression chambers of the at least two back wheel rams.

23. (cancelled)

24. A vehicle suspension system having a damping and stiffness system according to claim 6 further including a pressure maintenance device connected in fluid communication to at least four of the secondary chambers in the load distribution unit by respective pressure maintenance passages to maintain the static pressure of said at least four secondary chambers at a substantially common pressure.

25. A vehicle suspension system having a damping and stiffness system according to claim 24 further including a valve in each pressure maintenance passage.

26. A vehicle suspension system having a damping and stiffness system according to claim 24 further including a restriction in each pressure maintenance passage.

27. A vehicle suspension system having a damping and stiffness system as claimed in claim 24 wherein the pressure maintenance device includes a fluid pressure source.

28. A vehicle suspension system having a damping and stiffness system as claimed in claim 24 wherein the pressure maintenance device includes an accumulator.

29. A vehicle suspension system having a damping and stiffness system according to claim 27 wherein the pressure maintenance unit is controlled to regulate the static pressure in the at least four secondary chambers to a preset pressure.

30. A vehicle suspension system having a damping and stiffness system according to claim 29 wherein the preset pressure can be varied.

31. A vehicle suspension system having a damping and stiffness system according to claim 6 further including a pressure maintenance device, the pressure maintenance device including a first and a second output pressure, the first output pressure being connected to the first front, second front, first back and second back system chambers of the load distribution unit by respective system pressure maintenance passages, the second output pressure being connected to the front pitch volume and the back pitch volume by respective pitch pressure

maintenance passages.

32. A vehicle suspension system having a damping and stiffness system according to claim 31 wherein the pressure maintenance device includes a fluid pressure source, the pressure in the system chambers being controlled to a first preset pressure, the pressure in the pitch volumes being controlled to a second preset pressure, the first preset pressure being variable to vary the roll stiffness of the damping and stiffness system separately to the pitch stiffness, the second preset pressure being variable to vary the pitch stiffness of the damping and stiffness system.

33. A vehicle suspension system having a damping and stiffness system according to claim 2 further including resilient centering devices to provide a centering force on the piston rod assemblies in the load distribution unit to bias the piston rod assemblies towards a mid-stroke position.

34. A vehicle suspension system having a damping and stiffness system according to claim 7 further including a pressure maintenance device connected in fluid communication to at least four of the secondary chambers in the load distribution unit by respective pressure maintenance passages to maintain the static pressure of said at least four secondary chambers at a substantially common pressure.

35. A vehicle suspension system having a damping and stiffness system as claimed in claim 7 wherein the wheel rams of at least the two front or the two rear wheel rams are single-acting rams.

36. A vehicle suspension system having a damping and stiffness system according to claim 7 wherein each wheel ram is a double-acting ram further including a rebound chamber, the rebound chamber of each double-acting wheel ram being connected to the compression chamber of the diagonally opposite wheel ram.

APPENDIX B

Evidence: None

APPENDIX C

Related Proceedings: None

APPENDIX D

US006270098B1

(12) United States Patent
Heyring et al.

(10) **Patent No.: US 6,270,098 B1**

(45) **Date of Patent: Aug. 7, 2001**

(54) **LOAD DISTRIBUTION UNIT FOR VEHICLE SUSPENSION SYSTEM**

(75) Inventors: **Christopher Brian Heyring**, Eagle Bay; **Richard Monk**, Dunsborough, both of (AU)

(73) Assignee: **Kinetic Limited**, Dunsborough (AT)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

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PCT Pub. Date: **May 7, 1998**

(30) **Foreign Application Priority Data**

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Nov. 7, 1996	(AU)	PP 3505

(51) **Int. Cl.**^{7} **B60G 21/06**

(52) **U.S. Cl.** **280/124.161**; 280/124.104; 280/124.106; 280/6.155; 280/5.505; 280/5.507

(58) **Field of Search** ..... 280/124.104, 124.106, 280/124.161, 124.157, 124.16, 124.107, 124.1, DIG. 1, 6.155, 6.154, 5.506, 5.505, 5.507; 180/41

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Primary Examiner—Daniel G. DePumpo
Assistant Examiner—F. Zeender
(74) *Attorney, Agent, or Firm*—Birch, Stewart, Kolasch & Birch, LLP

(57) **ABSTRACT**

A load distribution unit is utilized in a vehicle suspension system having at least one pair of laterally adjacent forward wheel assemblies, and at least one pair of laterally adjacent rear wheel assemblies. A wheel ram is associated with each of the wheel assemblies, each wheel ram including a major chamber therein. The load distribution unit includes a plurality of fluid chambers, each fluid chamber being divided into at least two control chambers by at least one piston supported therein. Two pairs of the control chambers which vary in volume proportionally and in opposite senses therein with piston motion are system chambers, and at least two of the remaining control chambers are bump chambers. The pistons are interconnected by at least one connection device. The major chamber of each wheel ram is in fluid communication with a respective system chamber. The vehicle suspension system provides a roll stiffness and a pitch stiffness while providing minimum cross-axial articulation stiffness.

10 Claims, 6 Drawing Sheets

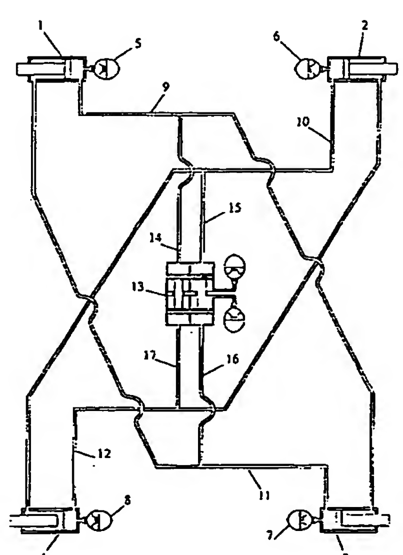

Fig 1.

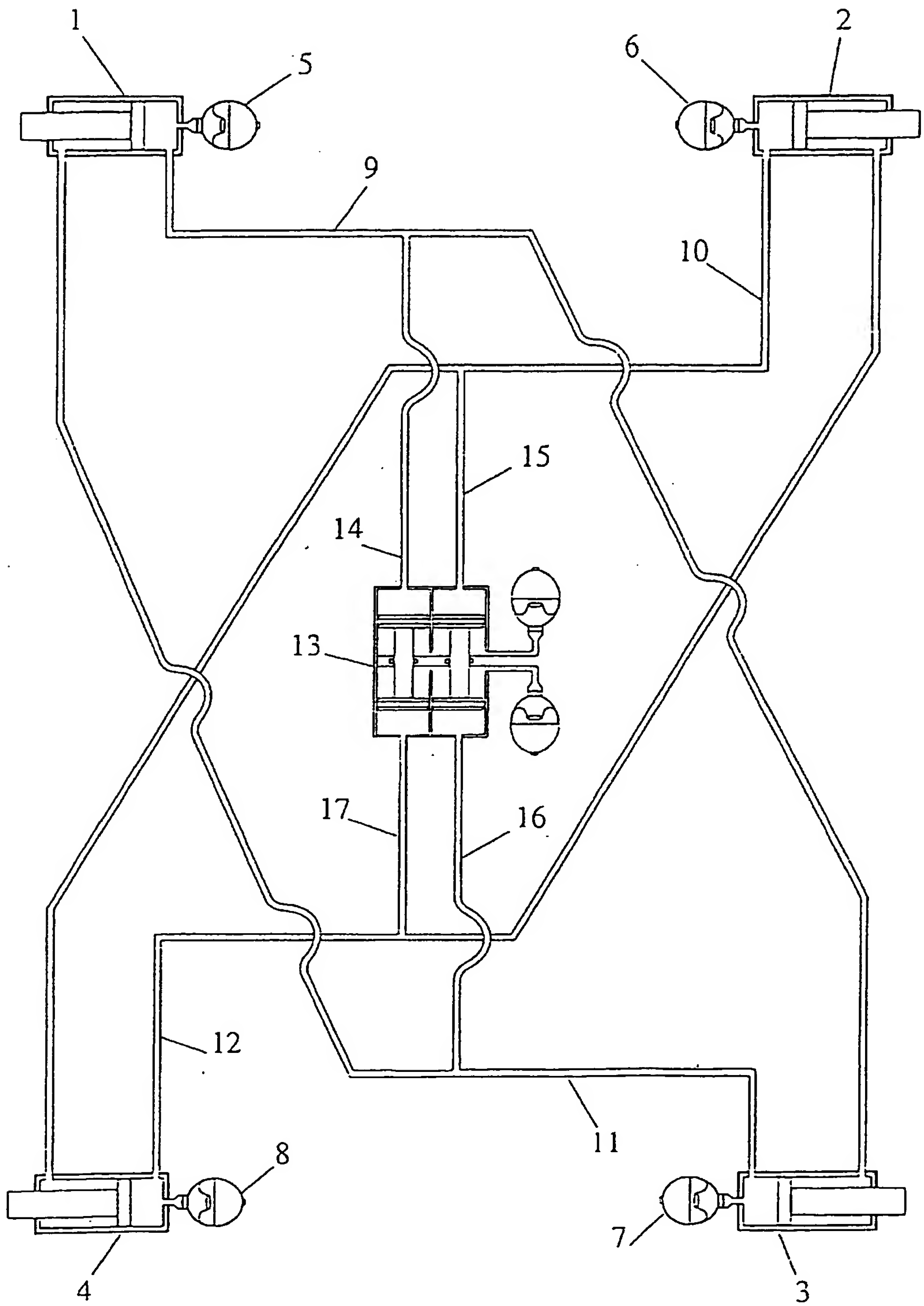

Fig 2.

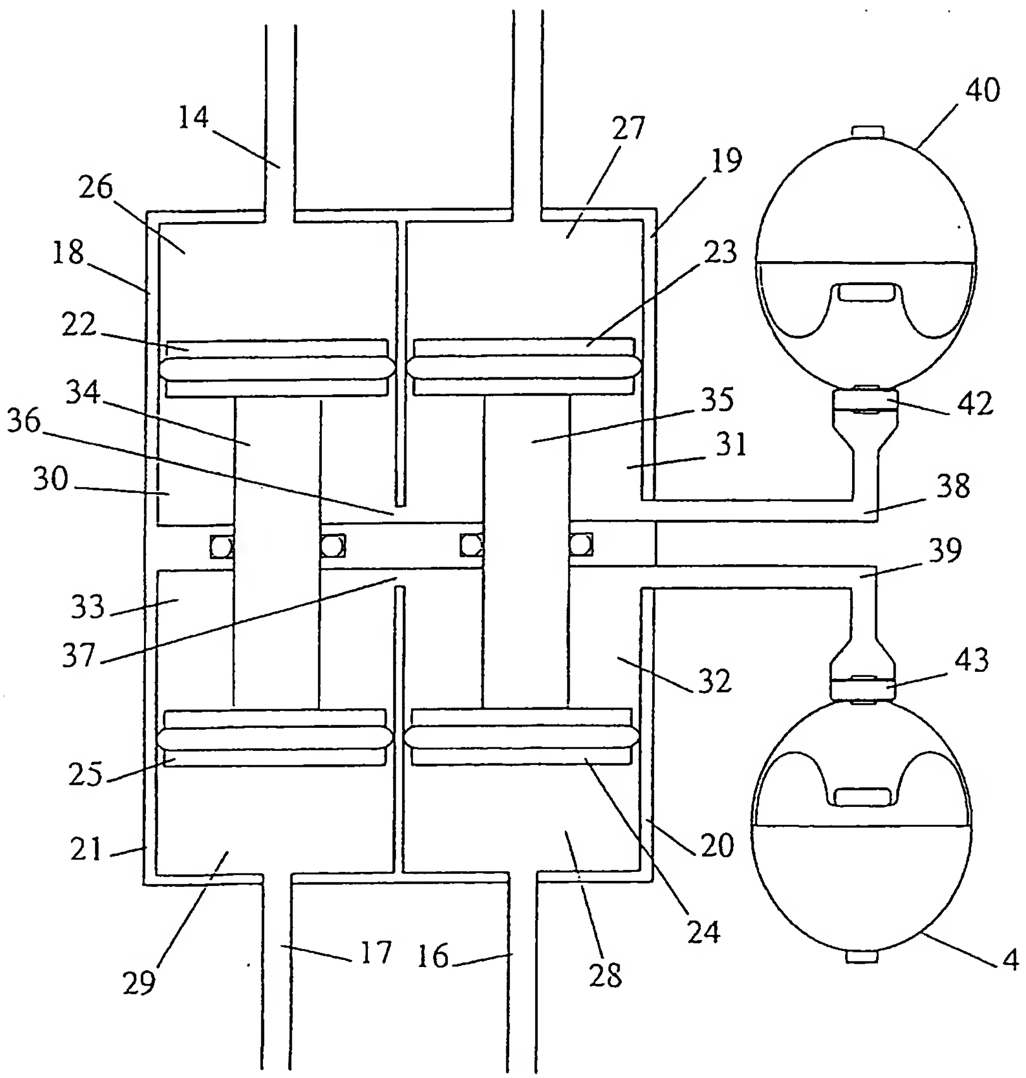
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Fig 3.

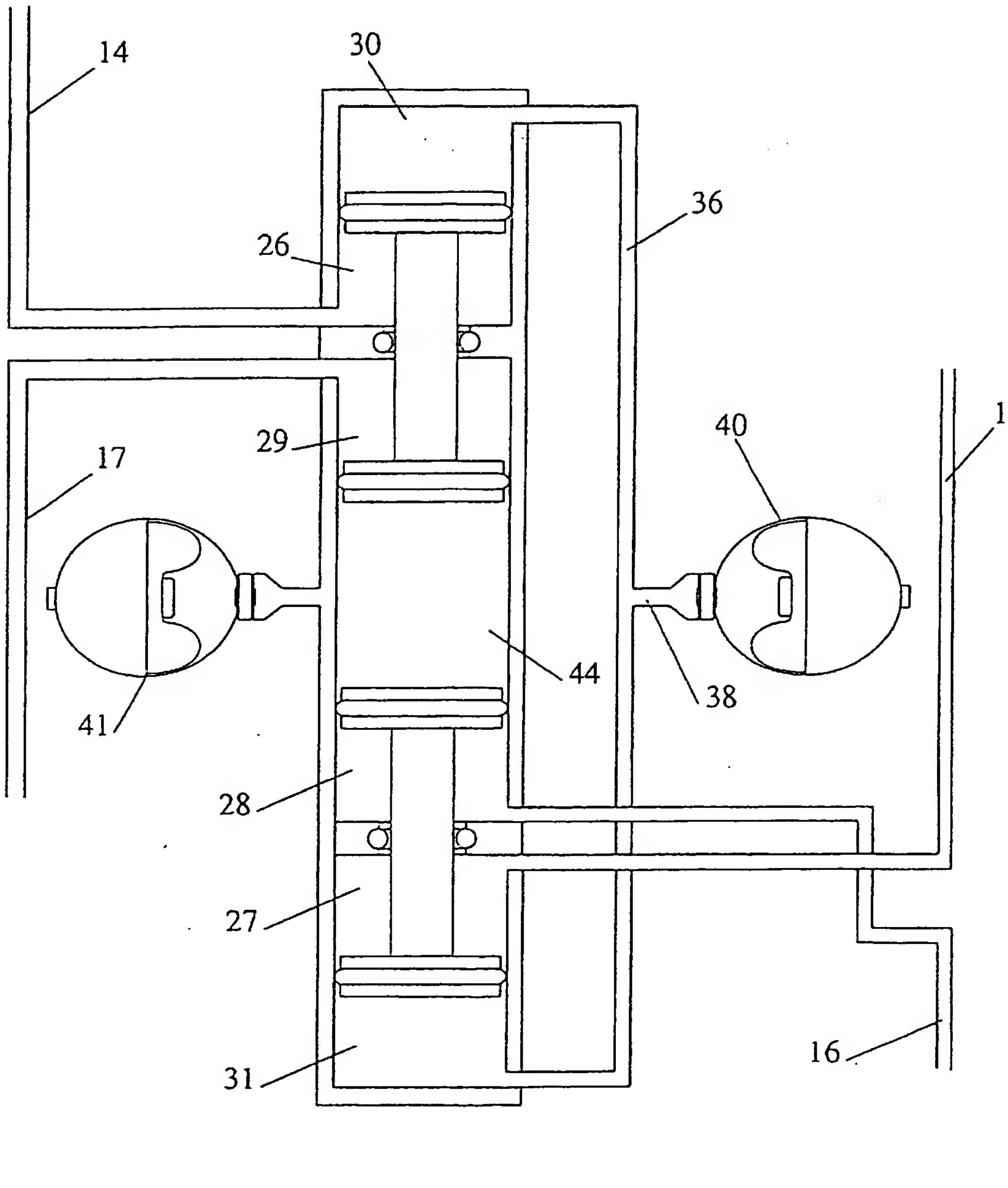

Fig 4.

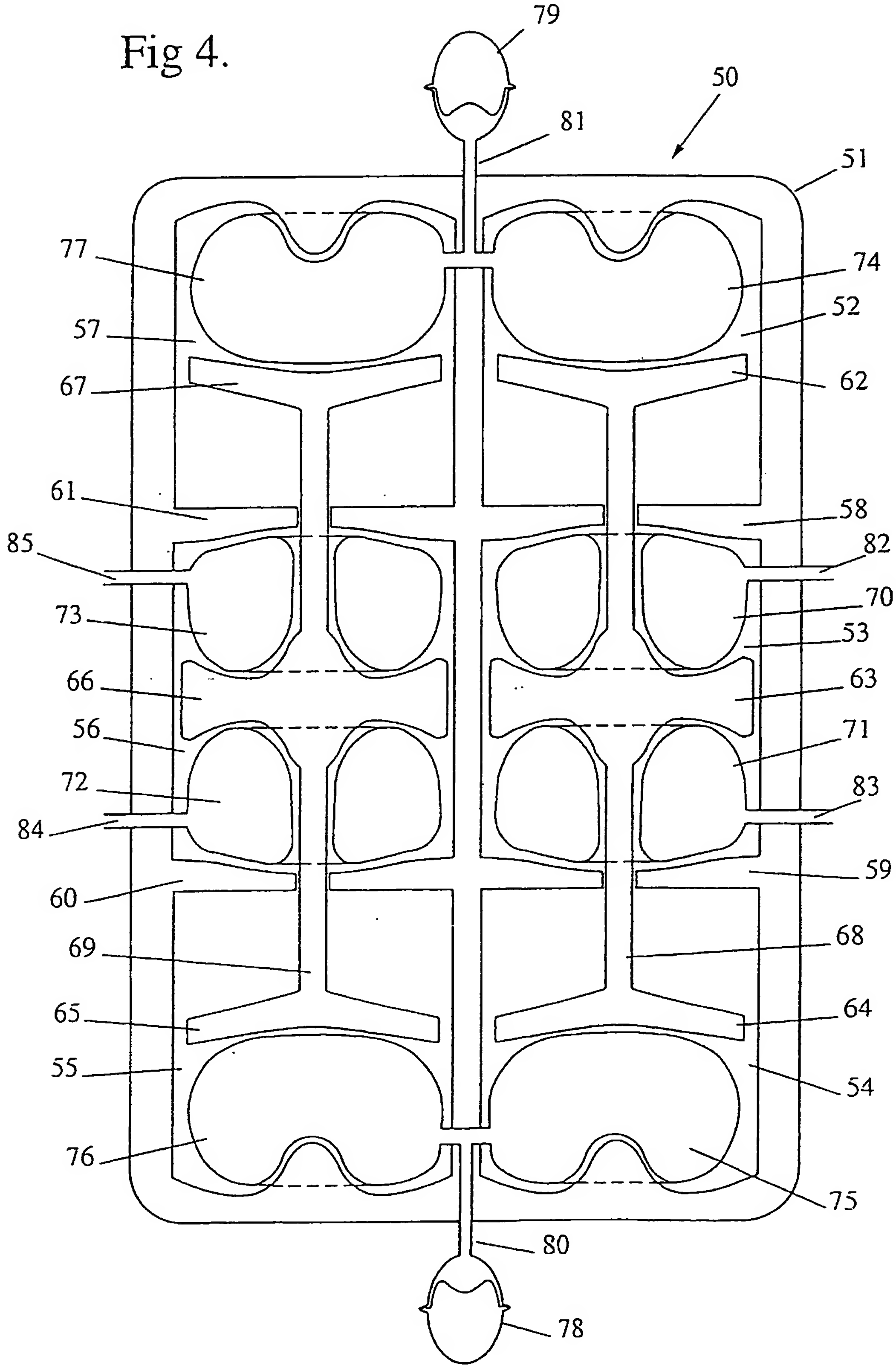

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Fig 5.

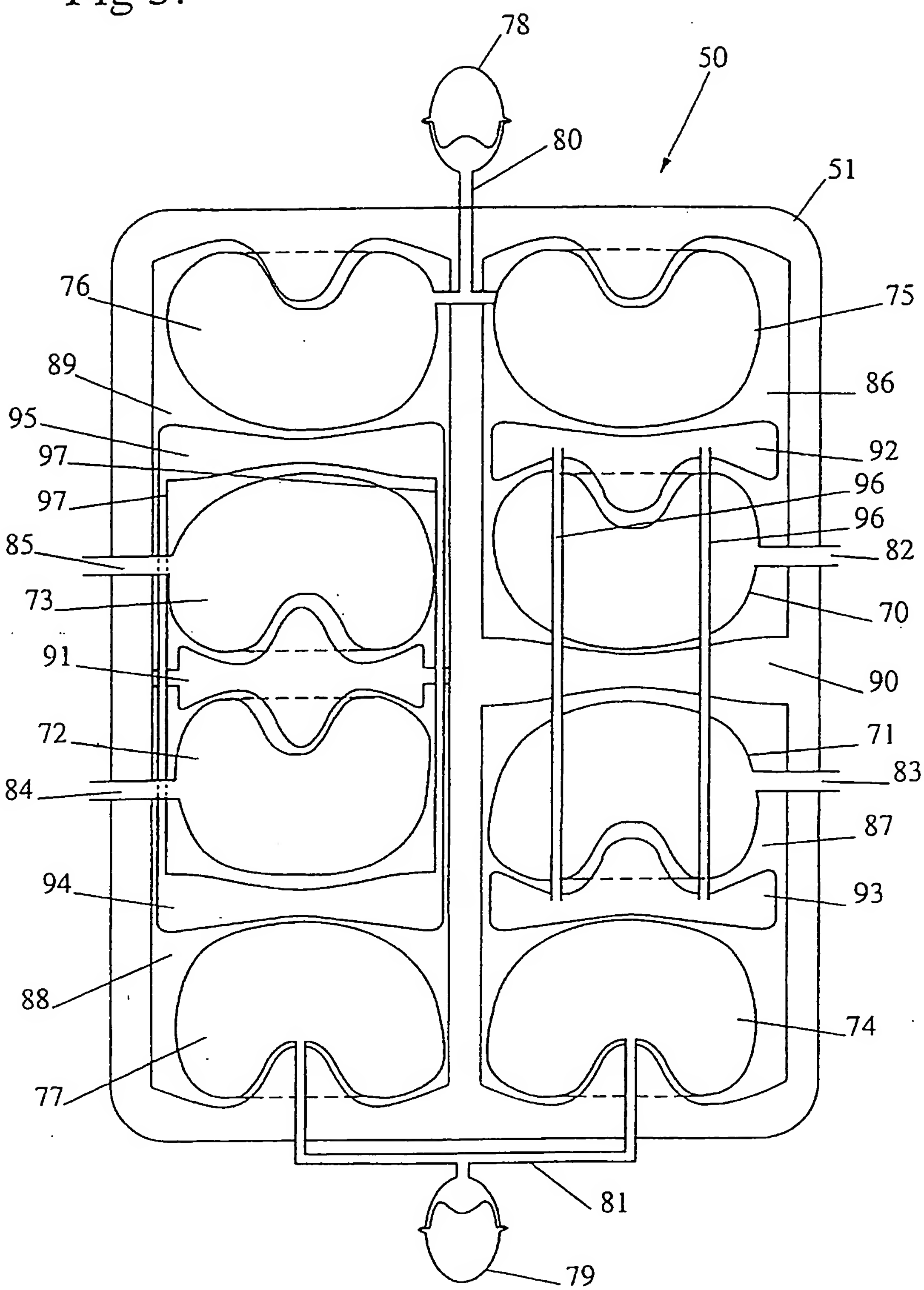
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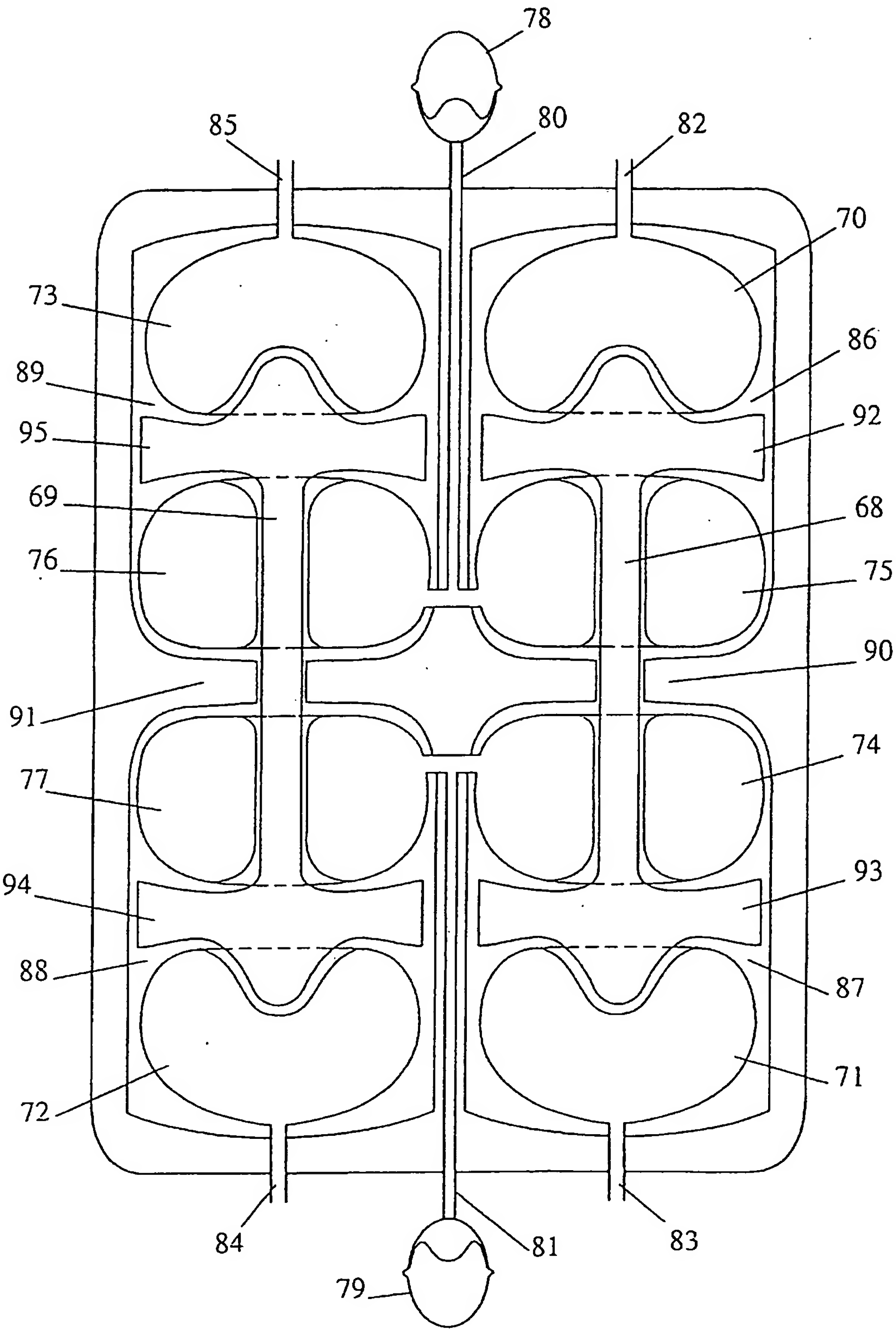
Fig 6.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LOAD DISTRIBUTION UNIT FOR VEHICLE SUSPENSION SYSTEM

This application is the national phase under 35 U.S.C. §371 of prior PCT International Application No. PCT/AU97/00719 which has an International filing date of Oct. 28, 1997 which designated the United States of America.

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention is generally directed to vehicle suspension systems. and in particular to a load distribution unit for a vehicle suspension system.

2. Description of the Background Art

The applicant has previously developed a vehicle fluid suspension system including a load distribution unit which performs the function of redistributing fluid between two pairs of diagonally interconnected double-acting or four single-acting rams respectively provided at each wheel assembly of the vehicle during cross-axle articulation motions, whilst opposing roll motions and introducing a controlled magnitude of pitch resilience. Such a suspension system is described in the applicant's International Application No. PCT/AU95/00096, details of which are incorporated herein by reference.

SUMMARY OF THE INVENTION

The present invention relates to an improved construction of said load distribution unit which can potentially reduce the packaging volume and weight of the original arrangement of the unit by more than 30% thereby increasing the mass production viability of the suspension system as a whole.

With this in mind, according to one aspect of the present invention, there is provided a load distribution unit for a vehicle suspension system having at least one pair of laterally adjacent forward wheel assemblies, and at least one pair of laterally adjacent rear wheel assemblies, a wheel ram associated with each said wheel assembly, each wheel ram including a major chamber therein,

wherein the load distribution unit includes two pairs of axially aligned fluid chambers, each fluid chamber being divided into two control chambers by a piston supported therein, in each pair of axially aligned fluid chambers two of the control chambers which vary in volume proportionally and in opposite senses therein with piston motion are system chambers, the remaining two chambers in each pair of axially aligned fluid chambers being bump chambers,

the piston of each said axially aligned pair of fluid chambers being interconnected by a connection means, the major chamber of each said wheel ram being in fluid communication with a respective said system chamber, such that as the vehicle suspension system provides a roll stiffness and a pitch stiffness while providing minimal cross-axle articulation stiffness,

and wherein a fluid communication is provided between pairs of the bump chambers such that the fluid pressure within the communicating bump chambers is transferable therebetween to thereby enable a pressure balance to be achieved between the system chambers.

The connection means interconnecting the pistons may be a rod member extending through the two control chambers in the middle of each pair of axially aligned fluid chambers.

A respective pair of the fluid chambers may be connected to the major chambers of the wheel rams on each side of the vehicle and the pistons located within each said respective pair of fluid chambers may be urged for movement in opposing axial directions to thereby enable the suspension system to resist roll motion by providing a roll stiffness while also providing a minimal cross-axle articulation stiffness.

Furthermore, the piston located within the pair of fluid chambers connected to the major chambers of the wheel rams at the front or rear of the vehicle may be urged for movement in opposing axial directions when the wheel assemblies are undergoing cross-axle articulation motion and may be urged for movement in the same axial direction when the vehicle is undergoing pitch motion to thereby confer minimal articulation stiffness and provide a pitch stiffness which is independent of the roll, four wheel bounce or articulation stiffnesses.

The bump chambers may be in fluid communication with accumulator means to thereby allow for a greater degree of resilience for the vehicle suspension system such that transient vertical motions of the wheel assemblies which can arise when the vehicle is travelling over a speed bump can be accommodated by the load distribution unit.

The fluid chambers may be of differing sizes to enable the pressures in the load distribution unit to be set as required during the design process. Each pair of fluid chambers may be located in parallel adjacent relation. Alternatively, each pair of chambers may be positioned in different positions in the vehicle or aligned along a common axis.

According to a second aspect of the invention there is provided a load distribution unit for a vehicle suspension system having at least one pair of laterally adjacent forward wheel assemblies, and at least one pair of laterally adjacent rear wheel assemblies, a wheel ram associated with each said wheel assembly, each wheel ram including a major chamber therein,

wherein the load distribution unit includes three fluid chambers aligned along a common axis to thereby provide opposing end chambers and a central chamber therebetween,

the end chambers being respectively divided by a piston supported therein into two control chambers, the central chamber being divided by two pistons into two control chambers and a central bump chamber,

two of the control chambers which vary in volume proportionally and in opposite senses with piston motion being separate bump chambers, the remaining four control chambers being system chambers,

respective connection means interconnecting each of the pistons in the central chamber to the piston in an adjacent said end chamber, the major chamber of each said wheel ram being in fluid communication with a respective one of the system chambers, such that the vehicle suspension system provides a roll stiffness and a pitch stiffness while providing minimal cross-axle articulation stiffness,

the two separate bump chambers of each end chamber being in fluid communication such that the fluid pressure within the communicating bump chambers is transferable therebetween to thereby enable a pressure balance to be achieved between the system chambers.

The connection means interconnecting the pistons may be a rod member extending through a said control chamber of the central chamber and a said control chamber of the end chamber adjacent thereto.

The two separate bump chambers may be in fluid communication with an accumulator means, and the central bump chamber may be in fluid communication with an accumulator means.

In devices such as rams and load distribution units described above, the problem of stationary friction or "stiction" where there is an initial resistance to movement of a stationary piston in a chamber can arise. This undesirable effect is especially prevalent in seals where there exists a large pressure difference across the seals which energises the seal firmly into the sealing surface giving high levels of friction. It is commonly found that there is only a certain reduction of the energising force possible (giving a set reduction in friction levels) whilst still maintaining a low fluid loss seal. This friction level can significantly retard the response time of the suspension system which can be detrimental to the ride comfort of the vehicle. The application seeks to overcome this problem by utilising fluid containers having at least a portion which is flexible to function as the chambers of the fluid ram. A similar problem can also arise in a load distribution unit with stiction between the piston seals and the bores, and the rod seals and the rods.

Hence, according to a further aspect of the present invention, there is provided a load distribution unit for a vehicle suspension system having at least one pair of laterally adjacent forward wheel assemblies, and at least one pair of laterally adjacent rear wheel assemblies, a wheel ram associated with each said wheel assembly, each wheel ram including a major chamber therein,

wherein the load distribution unit includes a housing divided into a pair of chamber sets, each chamber set including two axially aligned end chambers and a central chamber located and axially aligned therebetween,

pistons respectively located within the central chamber and within each said end chamber, the pistons being interconnected to thereby provide for common movement of the interconnected pistons therein, the piston: within the central chamber dividing said chamber into two system chambers, the piston within each said end chamber dividing said end chamber to provide a bump chamber on one side thereof,

a flexible fluid container being located within each said system chamber and being respectively in fluid communication with the major chamber of a said wheel ram such that the vehicle suspension system provides a roll stiffness and a pitch stiffness while providing minimal cross-axle articulation stiffness,

flexible fluid container being located within each bump chamber, with fluid communication being provided between the fluid containers in each pair of bump chambers such that the fluid pressure within the communicating bump chambers is transferable therebetween to thereby enable a pressure balance to be achieved between the system chambers.

Alternatively the load distribution unit may be arranged as described hereinafter, this alternate form being preferable from the manufacturing and packaging standpoints.

According to yet another aspect of the present invention there is provided a load distribution unit for a vehicle suspension system having at least one pair of laterally adjacent forward wheel assemblies, and at least one pair of laterally adjacent rear wheel assemblies, a wheel ram associated with each said wheel assembly, each wheel ram including a major chamber therein,

wherein the load distribution unit includes a housing divided into a pair of chamber sets, each chamber set including two axially aligned chambers,

pistons respectively located within each said chamber, the pistons being interconnected by a connection means to thereby provide for common movement of the interconnected pistons therein, each piston dividing its respective chamber into two control chambers, in each pair of axially aligned chambers, two of the control chambers which in volume are inversely proportional therein with piston motion are system chambers, the remaining two control chambers in each pair of axially aligned chambers being bump chambers,

a flexible fluid container being located within each said system chamber and being respectively in fluid communication with the major chamber of a said wheel ram such that the vehicle suspension system provides a roll stiffness and a pitch stiffness while providing minimal cross-axle articulation stiffness,

a flexible fluid container being located within each bump chamber, with fluid communication being provided between the fluid containers of each pair of bump chambers such that such that the fluid pressure within the communicating bump chambers is transferable therebetween to thereby enable a pressure balance to be achieved between the system chambers.

In the load distribution units as described above, the fluid containers located in the bump chambers and in fluid communication may also be in fluid communication with an accumulation means such as a hydropneumatic accumulator.

It is therefore generally possible to use the fluid containers in any of the load distribution units described above to replace the conventional hydropneumatic piston/chambers arrangement.

The load distribution unit may be used in a vehicle suspension system wherein the wheel ram is double acting having said major chamber and a minor chamber in which a piston rod of the wheel ram is located, the major chamber of each wheel ram being in direct fluid communication with the minor chamber of a diagonally opposite said wheel ram by a fluid communicating conduit, with each said system chamber of the load distribution unit being a fluid communication with a respective said fluid communicating conduit. Alternatively, the load distribution unit may be used in a vehicle suspension system wherein the wheel rams are single acting.

According to a further aspect of the present invention, there is provided a vehicle suspension system including a load distribution unit as described above.

The vehicle suspension system may be controlled by the control method described in the Applicant's International Application No: PCT/AU96/00397, details of which are incorporated herein by reference.

Further scope of applicability of the present invention will become apparent from the detailed description given hereinafter. However, it should be understood that the detailed description and specific examples, while indicating preferred embodiments of the invention, are given by way of illustration only, since various changes and modifications within the spirit and scope of the invention will become apparent to those skilled in the art from this detailed description.

BRIEF DESCRIPTION OF THE DRAWINGS

It will be convenient to further describe the invention with reference to the accompanying drawings which illustrate possible embodiments of a load distribution unit according to the present invention, although other arrangements are also envisaged. Consequently the particularity of the accompanying drawings is not to be understood as superseding the generality of the preceding description of the invention.

In the drawings:

FIG. 1 is a schematic view of a vehicle suspension system incorporating a preferred embodiment of a load distribution unit according to the present invention;

FIG. 2 is an enlarged schematic view of the load distribution unit of FIG. 1; and

FIGS. 3 to 6 are schematic views of other alternative preferred embodiments of a load distribution unit according to the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to FIG. 1 there are four double acting hydraulic rams (1, 2, 3, 4) shown interconnected between the vehicle body and the support means of the vehicle (for example, wheels, floats, skis). The layout corresponds to a plan view of the vehicle with the front being towards the top of the page, so the hydraulic ram 1 is associated with the front left support means of the vehicle and the hydraulic ram 4 is associated with the back left support means of the vehicle. Each ram has a hydropneumatic accumulator (5, 6, 7, 8) in fluid communication with the major chamber of the ram via a damper valve. The major chamber of each ram is in direct fluid communication with the minor chamber of the diagonally disposed ram by fluid communicating conduits (9, 10, 11, 12). The four fluid communicating conduits are connected to a load distribution unit 13 by respective branch lines (14, 15, 16, 17).

The same first embodiment of the load distribution unit 13 is shown in FIG. 2, enlarged for clarity, with the same reference numerals being used for common components.

The load distribution unit 13 comprises two pairs of fluid chambers 18, 21 and 19, 20, each fluid chamber in a pair being aligned along a common axis, the axis of the two pairs being parallel. Each fluid chamber is divided into two chambers by pistons 22, 23, 24, 25 forming a system chamber 26, 27, 28, 29 and a bump chamber 30, 31, 32, 33 in each fluid chamber, the pistons of each adjacent aligned fluid chamber are connected by rods 34, 35. The major chamber of the front left ram 1 is in fluid communication with the front left system chamber 26 of the load distribution unit via conduit 9 and branch line 14. Similarly the system chambers 27, 28, 29 respectively are associated with the major chambers of the front right, back right and back left hydraulic rams 2, 3, 4 respectively.

The bump chambers 30, 31 in the front rams 18, 19 of the load distribution unit are interconnected by a passage 36 and are generally described as front bump chambers since as the front wheels of the vehicle ride over a bump, fluid is displaced from the major chambers of the front rams into the front system chambers of the load distribution unit. This pushes the pistons 22 and 23 rearwards expelling fluid from the front bump chambers along the conduit 38 into the front bump accumulator 40. Since the rods 34, 35 join the rearward pistons 24, 25 to the forward pistons 22, 23, as the front wheels are riding over a bump and the forward pistons 22, 23 are pushed rearwards, fluid is expelled from the back system chambers 28, 29, extending the rear suspension rams. Fluid is also drawn into the back bump chambers 32, 33, which are interconnected by a passage 37, and connected to a back bump accumulator 41 via conduit 39. To damp this motion, damper valves 42, 43 may be provided in the mouths of the bump accumulators.

It is important to note that the locations of the chambers described above is only one of a number of connection arrangements possible with the above style of load distribution unit. For example the system chambers could all be located in the forward fluid chambers 18 and 19 giving a mechanical advantage between the front and rear systems which can be used to control roll split. The bump chambers are then accommodated within the remaining fluid chambers 20 and 21. Furthermore the fluid chambers which comprise the load distribution unit may be of differing diameters to increase the range of design options, along with the alternative chamber positions.

A second preferred embodiment of a load distribution unit according to the present invention is illustrated in FIG. 3. The branch lines (14, 15, 16, 17) to the system fluid conduits are shown in the same layout as in FIG. 1. For example the branch line 14 connects the front left system chamber 26 of the load distribution unit to the major chamber of the front left hydraulic suspension ram. The system chambers and bump chambers are swapped over compared to the earlier embodiments so that the system chambers are now the smaller volume chambers through which the rod passes, and the bump chambers are the larger chambers. One half of the load distribution unit has been rotated through one-hundred and eighty degrees and placed on one end of the other half, along a common axis. This places the back bump chambers together and they can be joined by removing the wall to make one common back bump chamber 44 and connected through conduit 39 to a back bump accumulator 41.

The two front bump chambers 30, 31 are located at the ends of the unit and interconnected by a passage 36, communicated with the front bump accumulator 40 via conduit 38. The function of the unit is similar to the first embodiment, the main difference being that the ratio of system to bump chamber areas can be reversed to give a wider range of sizing options to the designer.

The load distribution unit may also be used in a suspension system having single acting rams. In this arrangement, the major chamber of each ram can be in direct fluid communication with a system chamber of the load distribution unit.

FIG. 4 shows a straightforward application of fluid containers in the form of flexible bags to a third embodiment of a load distribution unit according to the present invention. The load distribution unit 50 comprises a housing 51 which is divided by dividing walls 58, 59, 60, 61 into six major chambers 52, 53, 54, 55, 56, 57 aligned along two parallel axes, three major chambers on each axis.

The major chamber in the centre on the left hand side of the figure is divided into two minor chambers by the central dividing piston 66, these minor chambers house individual system fluid bags 72 and 73 respectively. Similarly the major chamber in the centre on the right hand side of the figure is divided into two minor chambers by the central dividing piston 63, these minor chambers house individual system fluid bags 70 and 71 respectively. The system fluid bags are connected to the chambers of the actuators at each wheel by conduits 82, 83, 84, 85 in a connection sequence as described in the applicant's earlier noted patents and patent application so will not be further detailed herein. For the purposes of describing the operation of the present invention it will be assumed that the four system fluid bags communicate with the major chambers of the rams in corresponding positions, for example the left hand side forward system fluid bag 73 communicates with the major chamber of the ram associated with the front left wheel of the vehicle.

The left hand side forward major chamber 57 is divided by piston 67 forming two minor chambers, the most forward one of which accommodates a back bump fluid bag 77.

Similarly the right hand side forward major chamber **52** is divided by piston **62** forming two minor chambers, the most forward one of which accommodates the other back bump fluid bag **74**. The conduit **81** joining the two back bump fluid bags and the back bump accumulator **79** permits fluid flow between the bags and from the bags into the back bump accumulator.

Similarly the rearward major chambers **54, 55** contain dividing pistons **64, 65** respectively and the front bump bags **75, 76** respectively. The front bump bags are joined to each other and to the front bump accumulator **78** by the conduit **80**.

All three pistons **65, 66, 67** in the major chambers **55, 56, 57** on the left hand side of the unit **50** are joined together by the piston rod **69**. Likewise the pistons **62, 63, 64** in the major chambers on the right hand side are joined together by piston rod **68**.

When the front wheels of the vehicle ride over a bump and the corresponding actuators become compressed, fluid is expelled from the wheel rams into the associated system fluid bags **70, 73** in the load distribution unit. This causes the piston rods **68, 69** to be thrust rearwards, compressing the front bump fluid bags **75, 76** and forcing fluid into the associated front bump accumulator **78**.

A toroidal fluid bag may alternatively be placed in each free minor chamber, replacing the larger bump bags **74, 75, 76, 77** illustrated. It should be understood that if this is done, all bump bags must be replaced in a similar manner to retain the functionality of the load distribution unit. Also the new toroidal fluid bags next to the dividing walls **58** and **61** are now front bump bags replacing the illustrated front bump bags **75, 76** at the other end of the unit.

Similarly, the new toroidal fluid bags next to the dividing walls **59** and **60** are now back bump bags replacing the illustrated back bump bags **74, 77** at the opposite end of the unit.

FIG. **5** illustrates a fourth preferred embodiment of the load distribution unit **50** according to another aspect of the present invention. The essential functionality of the load distribution unit is not altered, yet the packaging length required is much reduced as only two aligned major chambers **86, 87** and **88, 89** are necessary on each side. The major chambers on the left hand side of the housing **51** are formed by the fixed dividing wall **91** which is shaped very like the central piston **66** in FIG. **4**. The forward major chamber **89** is divided into two minor chambers by the piston **95**. Each of these minor chambers houses a fluid bag **73, 76**. The front left system fluid bag **73** is connected to the front left wheel actuator as previously described for FIG. **4**, and is now positioned between the dividing wall **91** and the front left load distribution unit piston **95**. The minor chamber on the other side of the piston **95** contains a front bump fluid bag **76**. The rearward left hand side major chamber **88** is divided by a back left load distribution unit piston **94** into two minor chambers housing the back left system fluid bag **72** and a back bump fluid bag **77**. The two left hand load distribution unit pistons **94, 95** are fixed together by bars **97** arranged around the periphery of the pistons. The back left system fluid bag **72** is housed in the minor chamber between the dividing wall **91** and the back left load distribution unit piston **94** such that as the pistons move in unison, the volume of fluid in the back left system fluid bag **72** varies substantially reciprocally with the volume of fluid in the front left system fluid bag **73**.

The construction illustrated in FIG. **5** for the right hand side of the load distribution unit is similar to that for the left hand side, the changes being restricted to the shaping of the dividing wall **90** forming major chambers **86, 87**, and the shaping of the pistons **92, 93** in said chambers. The positioning of the fluid bags corresponds to the left hand side, so for example the front right system fluid bag **70** is housed in the minor chambers formed between the front right load distribution unit piston **92** and the dividing wall **90**. On the other side of the piston **92** is a front bump fluid bag **75**, which is connected to the front bump fluid bag **76** on the left hand side and the front bump accumulator **78** by the conduit **80**. The back right major chamber **87** is divided by the back right load distribution unit piston **93** into two minor chambers which house the back right system fluid bag **71** and a back bump fluid bag **74**. The front and back right load distribution unit pistons **92, 93** are fixed together by bars **96** arranged around the periphery of the pistons. The back right system fluid bag **71** is housed in the minor chamber between the dividing wall **90** and the back right load distribution unit piston **93** such that as the pistons move in unison, the volume of fluid in the back right system fluid bag **71** varies substantially reciprocally with the volume of fluid in the front right system fluid bag **70**. The back bump fluid bags **74, 77** and the back bump accumulator **79** are connected by the conduit **81**.

FIG. **6** shows a load distribution unit **50** of similar form to that illustrated in FIG. **5**, the differences being largely restricted to the positioning of the bump and system fluid bags. The load distribution unit pistons. are also now fixed together by piston rods **68, 69** as in FIG. **4**. This alternative arrangement allows the matching of pressures, areas and resultant forces to enable optimal sizing of the components during system design. The system bags **70, 71, 72, 73** now occupy the outermost minor chambers and the bump bags **74, 75, 76, 77** occupy the minor chambers on either side of the dividing walls **90, 91**. For example the front left major chamber **89** is divided by the front left load distribution unit piston **95** into two minor chambers, the outermost of which contains the front left system fluid bag **73**. The other minor chamber between the front left load distribution unit piston **95** and the dividing wall **91** houses a front bump fluid bag **76** which is connected to the other front bump fluid bag **75** and the front bump accumulator by the conduit **80** as described for the preceding embodiments of the load distribution unit according to the present invention.

It should be further noted that the major chambers may be of differing volumes, the piston rods **68, 69** may be extended through the ends of the casing and the end portions have two major chambers at one end of the housing **51**, and the front and back bump fluid bags may be housed in the two major chambers at the other end of the housing. Any or all of the above options can be used to assist in the matching of pressures, areas and resultant forces to enable optimal sizing of the components during system design.

Furthermore, the major chambers of the load distribution unit may be aligned along a single common axis as described in the applicant's prior patents and patent applications. This can be achieved by, for example, rotating the left hand side portion of the load distribution unit through 180° in plan view, then fixing it to either end of the right hand side portion. One of the bump fluid bags can then be discarded.

It is also envisaged that the load distribution unit be provided as two separate housings respectively controlling the left and right sides of the vehicle. These housings can then be positioned in separate locations within the vehicle.

The invention being thus described, it will be obvious that the same may be varied in many ways. Such variations are

not to be regarded as a departure from the spirit and scope of the invention, and all such modifications as would be obvious to one skilled in the art are to be included within the scope of the following claims.

The claims defining the invention are as follows:

1. A load distribution unit for a vehicle suspension system having at least one pair of laterally adjacent forward wheel assemblies, and at least one pair of laterally adjacent rear wheel assemblies, a wheel ram associated with each said wheel assembly, each wheel ram including a major chamber therein,

wherein the load distribution unit comprises a first pair of axially aligned fluid chambers, and a second pair of axially aligned fluid chambers, each fluid chamber being divided into two control chambers by a piston supported therein, the pistons in the first fluid chamber pair being interconnected by a first connection means and the pistons in the second fluid chamber pair being interconnected by a second connection means, said first and second connection means respectively comprising a rod extending through the two control chambers in the middle of each pair of axially aligned fluid chambers and contained entirely within respective said first and second pair of fluid chambers,

two of the control chambers in each fluid chamber pair being system chambers which vary in volume proportionally and in opposite senses therein with piston motion, the remaining two control chambers in each fluid chamber pair being bump chambers,

the system chambers of the first fluid chamber pair being in fluid communication with the major chambers of the wheel rams on one side of the vehicle, the system chambers of the second fluid chamber pair being in fluid communication with the major chambers of the wheel rams on the opposite side of the vehicle,

wherein in each fluid chamber pair, one system chamber is a front system chamber connectable to at least one of the forward wheel assemblies, the other system chamber is a rear system chamber connectable to at least one of the rear wheel assemblies on the same side of the vehicle, such that a pressure increase in the major chambers of the wheel rams on one side of the vehicle or a pressure decrease in the major chambers of the wheel rams on one side of the vehicle is reacted by the interconnection means, thereby providing a roll stiffness,

and wherein in each fluid chamber pair, one bump chamber is a front bump chamber which varies in volume proportionally and in an opposite sense to the front system chamber therein with piston motion, the other bump chamber being a rear bump chamber, to thereby provide pitch stiffness,

the front bump chambers of the first and second fluid chamber pairs being in fluid communication and the rear bump chambers of the first and second fluid chamber pairs being in fluid communication, such that fluid can be transferable therebetween to thereby provide a minimal cross-axle articulation stiffness,

the front bump chambers being in fluid communication with a front bump accumulator means and the rear bump chambers being in fluid communication with a rear bump accumulator means.

2. The load distribution unit according to claim 1, wherein the wheel ram is a double acting ram further comprising a minor chamber in which a piston rod of the wheel ram is located, the major chamber of each wheel ram being in direct fluid communication with the minor chamber of a diagonally opposite said wheel ram by a fluid communicating conduit, with each said system chamber of the load distribution unit being in fluid communication with a respective said fluid communicating conduit.

3. The load distribution unit according to claim 1, wherein a flexible fluid container is located within each said system chamber and is respectively in fluid communication with the major chamber of a said wheel ram, and a flexible fluid container is located within each bump chamber, with fluid communication being provided between the fluid containers in each pair of bump chambers such that the fluid pressure within the communicating bump chambers is transferable therebetween to thereby enable a pressure balance to be achieved between the system chambers.

4. The load distribution unit according to claim 1, wherein the fluid chamber pairs are joined together side by side such that the axis of the two fluid chambers of the first pair thereof is parallel to the axis of the two fluid chambers of the second pair thereof.

5. The load distribution unit according to claim 1, wherein the first and second fluid chamber pairs are located in different locations in the vehicle.

6. The load distribution unit according to claim 1, wherein the first and second fluid chamber pairs are axially aligned such that the two fluid chambers of the first pair and the two fluid chambers of the second pair are all substantially axially aligned along a common axis.

7. A load distribution unit for a vehicle suspension system having at least one pair of laterally adjacent forward wheel assemblies, and at least one pair of laterally adjacent rear wheel assemblies, a wheel ram associated with each said wheel assembly, each wheel ram including a major chamber therein,

wherein the load distribution unit includes three fluid chambers aligned along a common axis to thereby provide opposing first and second end chambers and a central chamber therebetween,

the first and second end chambers being respectively divided by a piston supported therein into two control chambers, the central chamber being divided by two pistons into two control chambers and a central bump chamber,

respective connection means interconnecting each of the pistons in the central chamber to the piston in an adjacent said end chamber to respectively provide first and second piston assemblies, said respective connection means comprising a rod extending between said first and second end chamber respectively and said central chamber, and contained entirely within said fluid chambers,

two of the control chambers providing first and second bump chambers, the remaining four control chambers providing system chambers,

wherein the two system chambers formed by each of the first and second piston assemblies, respectively provide front and rear system chambers respectively connectable to the major chamber of the wheel ram of a said forward wheel assembly, and a said rear wheel assembly on the same side of the vehicle, such that a pressure increase in the major chambers of the wheel rams on one side of the vehicle or a pressure decrease in the major chambers of the wheel rams on one side of the vehicle is reacted by the interconnection means, thereby providing a roll stiffness,

the central bump volume providing a first bump volume, the first and second bump chambers being in fluid

communication to form a second bump volume, such that when the wheel assemblies are undergoing cross-axle articulation motions, the first and second piston assemblies are urged to move in the same axial direction, thereby providing minimal cross-axle articulation stiffness,

the first and second bump volumes varying in volume proportionately and in opposite senses with relative motion between the first and second piston assemblies, one of said bump volumes providing a front bump volume which varies in volume proportionally and in an opposite sense to the front system chambers, the other said bump volume providing a rear bump volume, to thereby provide a pitch stiffness,

wherein the front bump volume is in fluid communication with a front bump accumulator means and the rear bump volume is in fluid communication with a rear bump accumulator means.

8. The load distribution unit according to claim 7, wherein the wheel ram is a double acting ram further comprising a minor chamber in which a piston rod of the wheel ram is located, the major chamber of each wheel ram being in direct fluid communication with the minor chamber of a diagonally opposite said wheel ram by a fluid communicating conduit, with each said system chamber of the load distribution unit being in fluid communication with a respective said fluid communicating conduit.

9. The load distribution unit according to claim 7, wherein a flexible fluid container is located within each said system chamber and is respectively in fluid communication with the major chamber of a said wheel ram, and a flexible fluid container is located within each bump chamber, with fluid communication being provided between the fluid containers located within the first and second bump chambers such that the fluid pressure within the communicating bump chambers is transferable therebetween to thereby enable a pressure balance to be achieved between the system chambers.

10. A load distribution unit for a vehicle suspension system having at least one pair of laterally adjacent forward wheel assemblies, and at least one pair of laterally adjacent rear wheel assemblies, a wheel ram associated with each said wheel assembly, each wheel ram including a major chamber therein,

wherein the load distribution unit includes a housing divided into a pair of chamber sets, each chamber set including two axially aligned end chambers and a central chamber located and axially aligned therebetween,

pistons respectively located within the central chamber and within each said end chamber, the pistons being interconnected to thereby provide for common movement of the interconnected pistons within each chamber set, the piston within the central chamber dividing said central chamber into two control chambers, the piston within each said end chamber dividing said end chamber to provide a control chamber on one side thereof, the piston interconnection means comprising a rod contained entirely within said housing,

two pairs of the control chambers providing bump chambers which vary in volume proportionally and in opposite senses with piston motion, the remaining two pairs of the control chambers providing system chambers,

a flexible fluid container being located within each said system chamber and being respectively adapted to be in fluid communication with the major chamber of a said wheel ram the interconnected pistons within each chamber set capable of being urged for movement in the same axial direction when the wheel assemblies are undergoing cross-axle articulation motion and capable of being urged for movement in opposing axial directions when the vehicle is undergoing pitch motion to thereby confer minimal articulation stiffness and provide a pitch stiffness which is independent of the roll, four wheel bounce or articulation stiffnesses,

a flexible fluid container being located within each bump chamber, with fluid communication being provided between the fluid containers in each pair of bump chambers such that the fluid pressure within the communicating bump chambers is transferable therebetween to thereby enable a pressure balance to be achieved between the system chambers,

wherein each said pair of bump chambers is in fluid communication with a bump accumulator means.

* * * * *

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Heyring et al.

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(45) Date of Patent: Jul. 13, 2004

(54) PASSIVE RIDE CONTROL FOR A VEHICLE SUSPENSION SYSTEM

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(73) Assignee: Kinetic Pty. Ltd., Dunsborough (AU)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

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PCT Pub. Date: Oct. 19, 2000

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(51) Int. Cl.[7] B60G 9/04
(52) U.S. Cl. 280/124.157; 280/5.507; 280/5.508; 280/124.159; 701/37
(58) Field of Search 280/124.157, 124.159, 280/124.106, 5.502, 5.507, 5.508; 701/37

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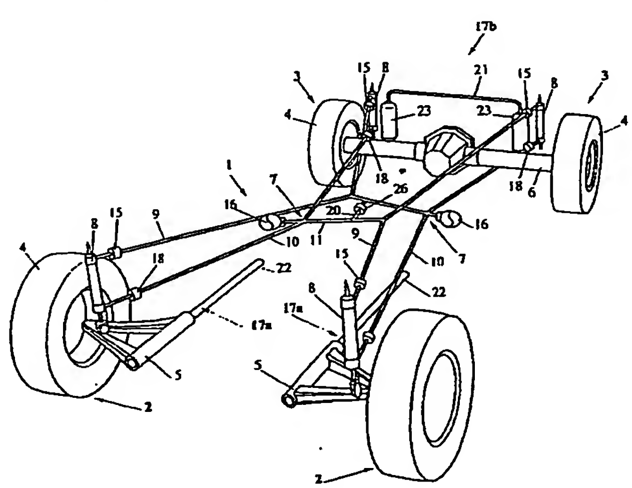
(57) **ABSTRACT**

A roll control system for a vehicle suspension system and a method for controlling the control system is disclosed. The vehicle has at least one pair of laterally spaced front wheel assemblies and at least one pair of laterally spaced rear wheel assemblies. Each wheel assembly includes a wheel and a wheel mounting permitting wheel movement in a generally vertical direction relative to the vehicle body, and vehicle support means for providing at least substantially a major portion of the support for the vehicle. The roll control system includes: wheel cylinders respectively locatable between each wheel mounting and the vehicle body. Each wheel cylinder includes an inner volume separated into first and second chambers by a piston supported within, and first and second fluid circuits respectively providing fluid connection between the wheel cylinders by fluid conduits. Each of the fluid circuits provide fluid communication between the first chambers on one side of the vehicle and the said second chambers on the opposite side of the vehicle to thereby provide roll support decoupled from a warp mode of the vehicle suspension system by providing a roll stiffness about a level roll attitude whilst simultaneously providing substantially zero warp stiffness. The method includes bypassing fluid flow from at least a substantial portion of the conduits during predetermined wheel inputs to the control system to thereby minimize line damping and/or fluid inertia effects on the damping of the control system.

30 Claims, 22 Drawing Sheets



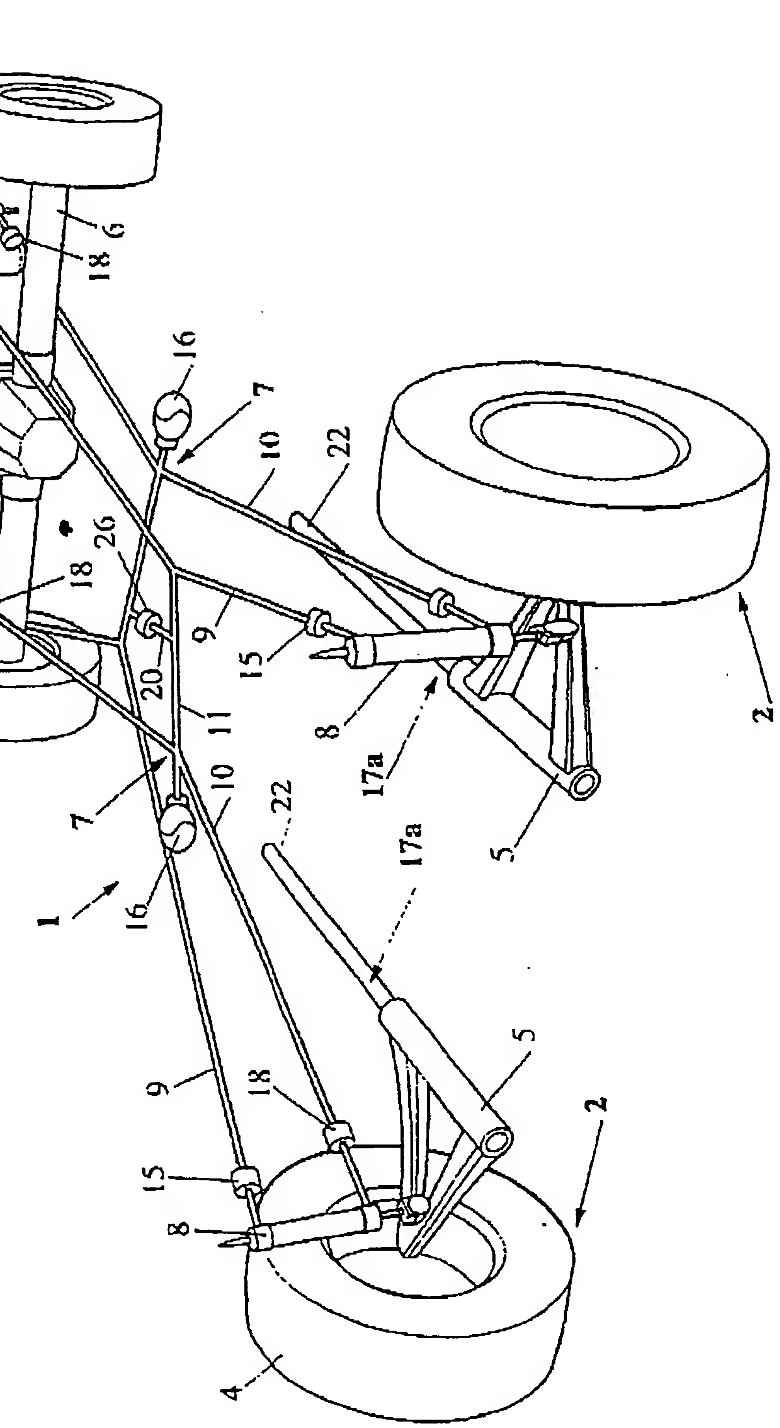


Fig 1.

Fig 2.

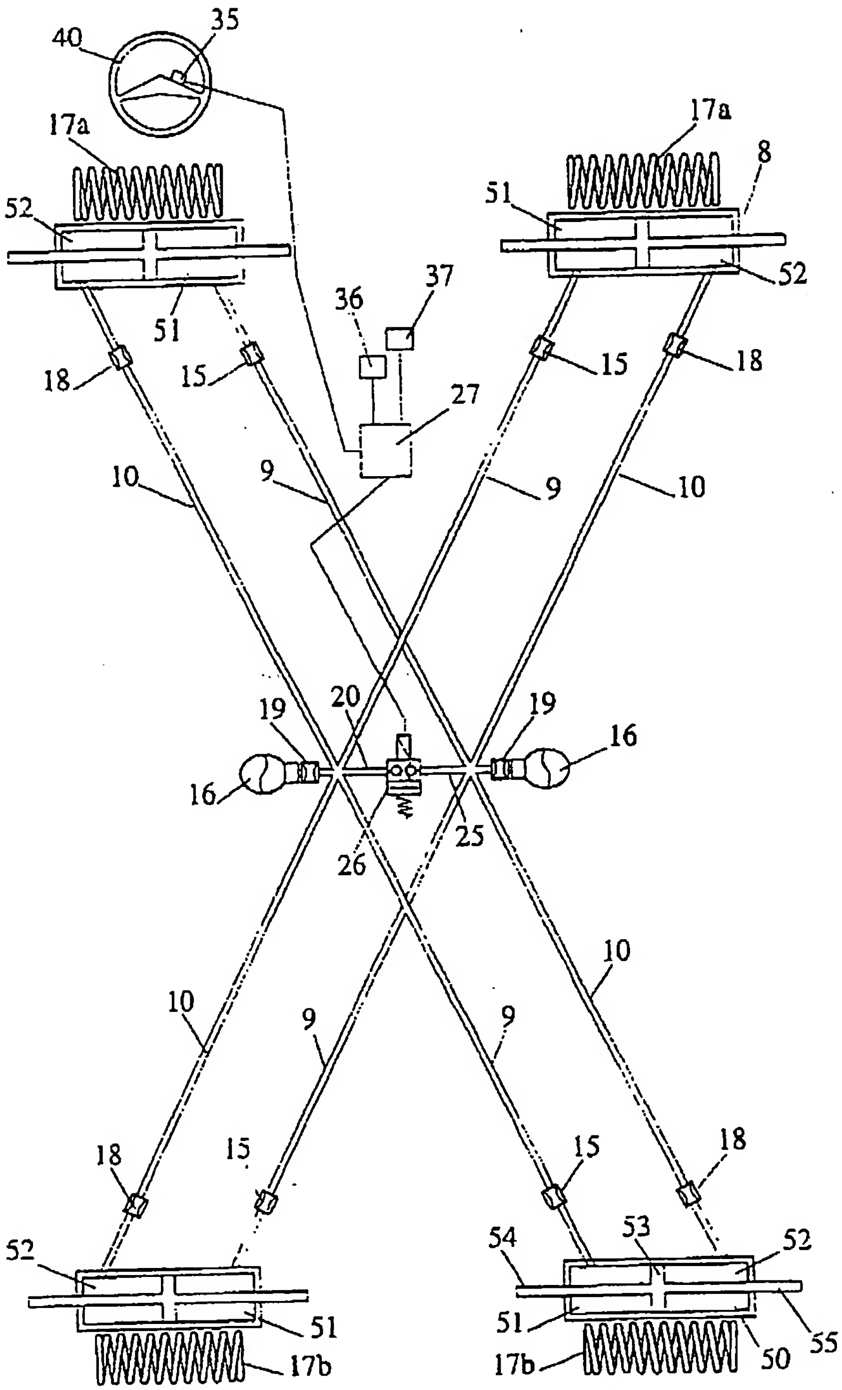

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Fig 3.

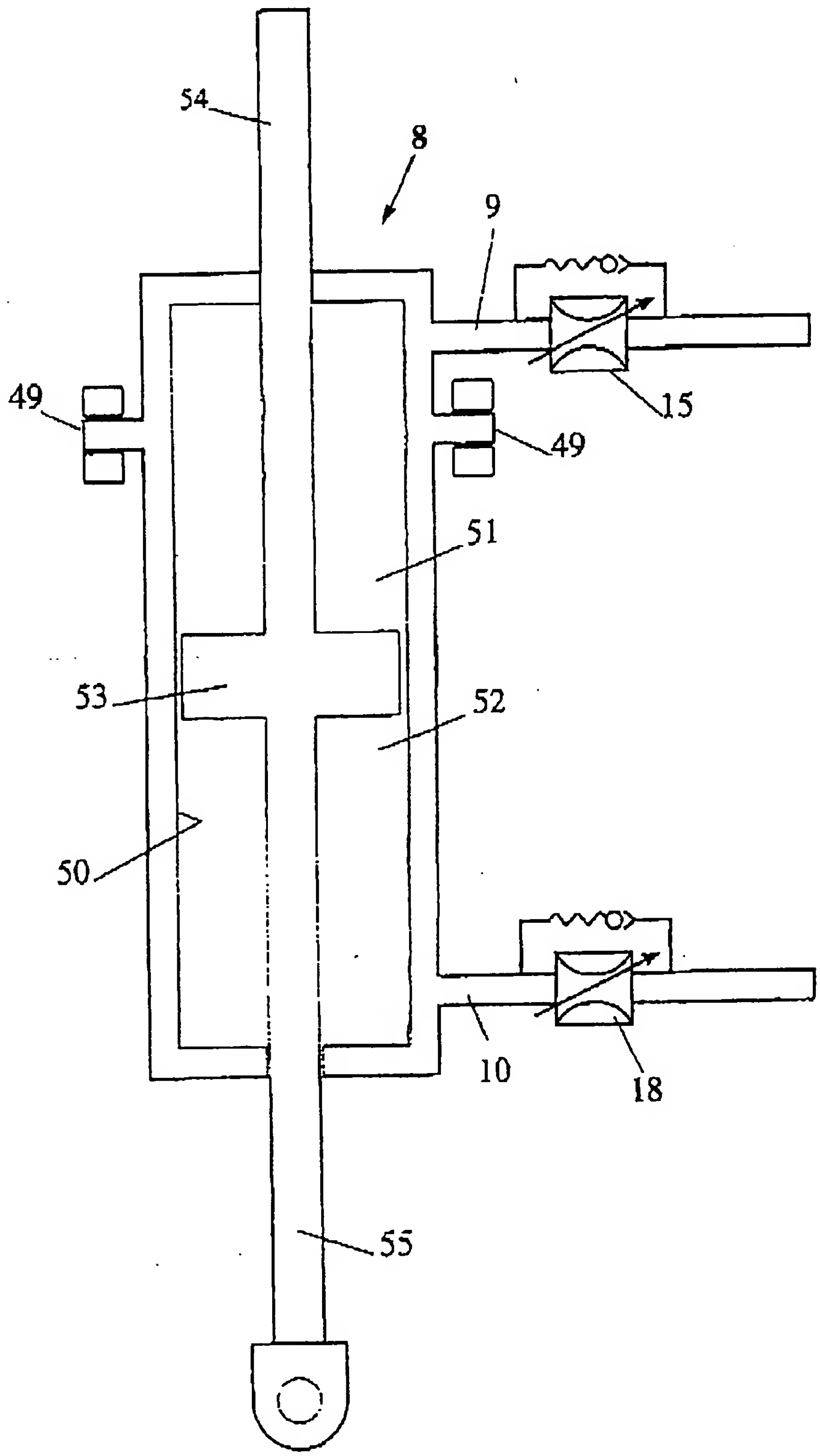


Fig 4.

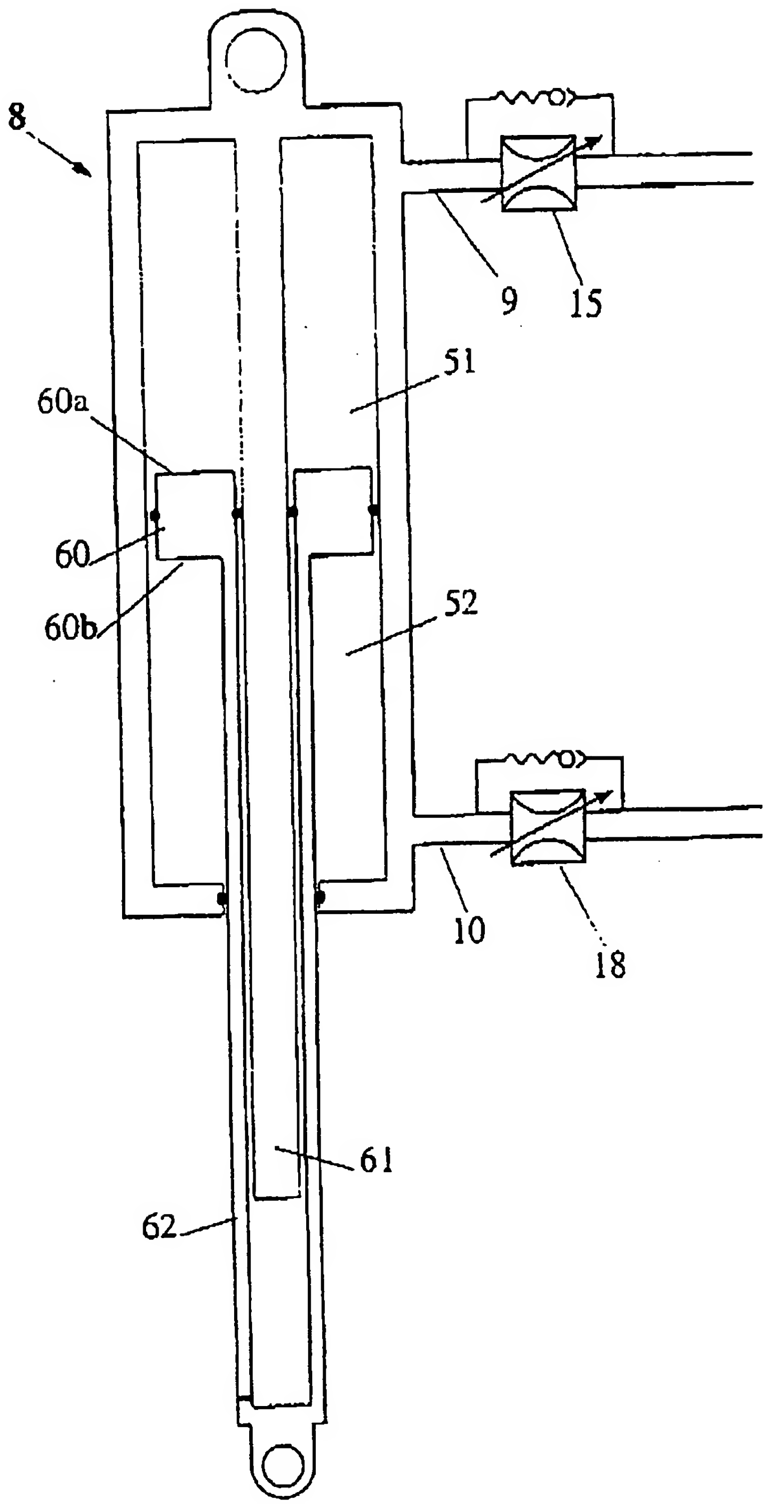
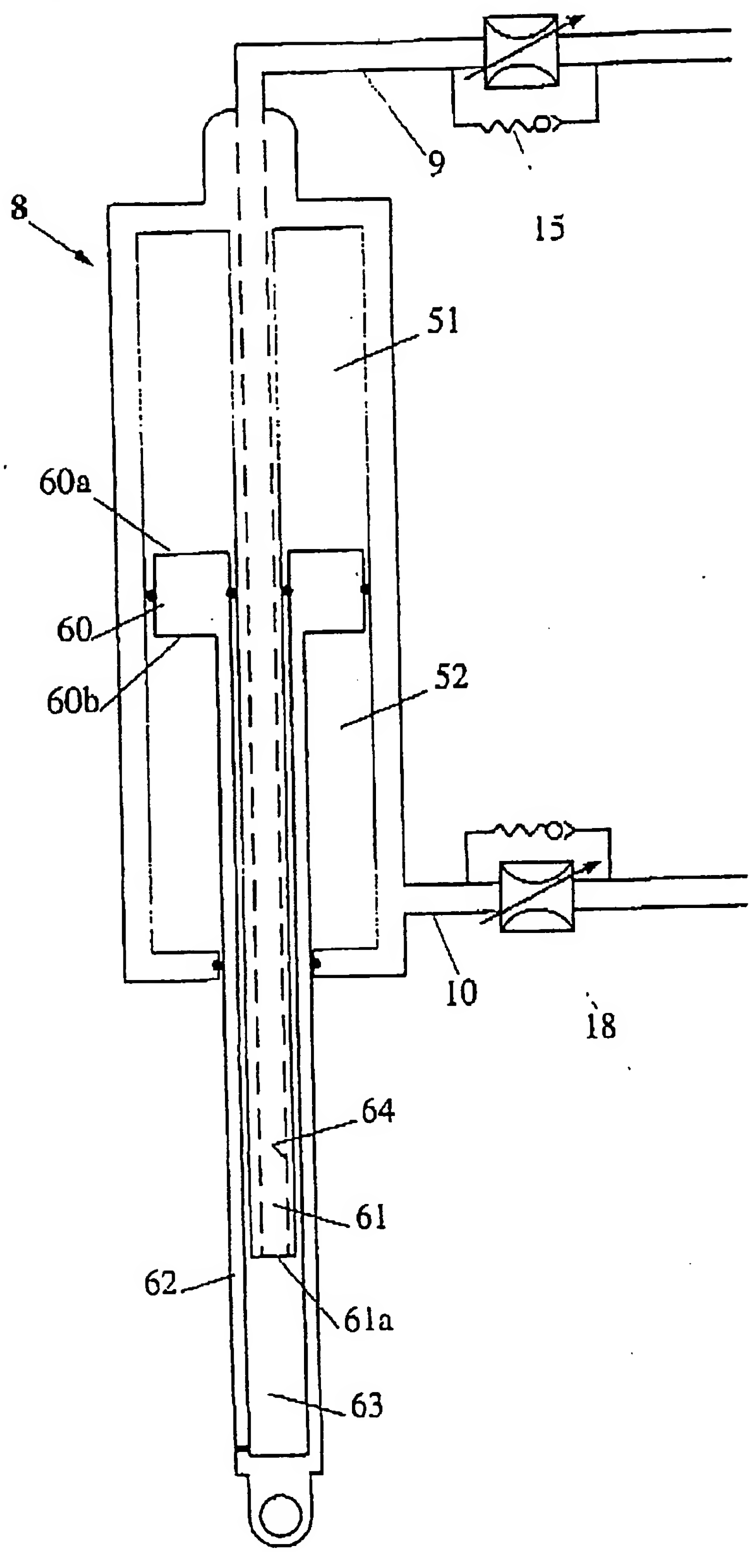


Fig 5.



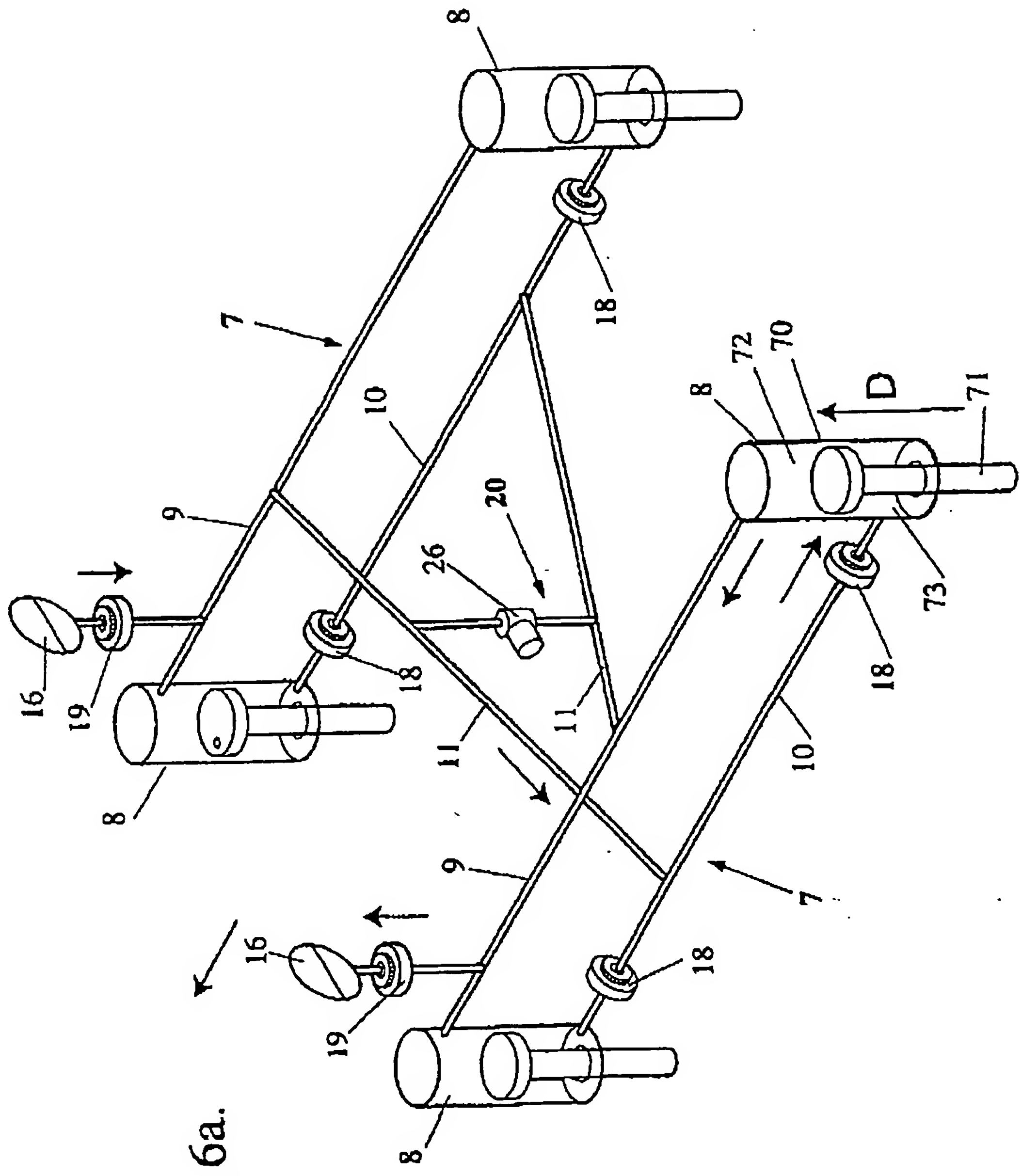


Fig 6a.

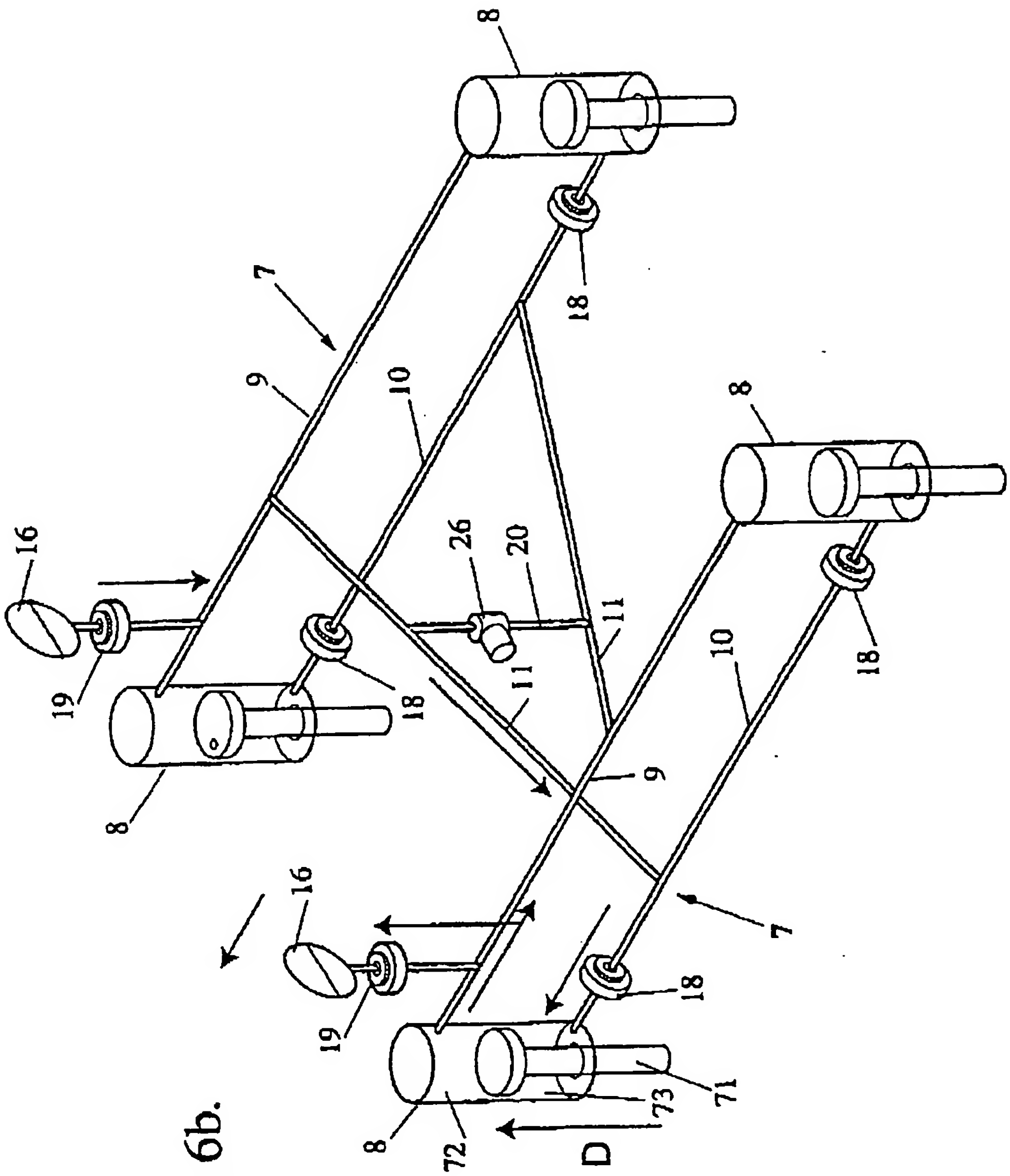


Fig 6b.

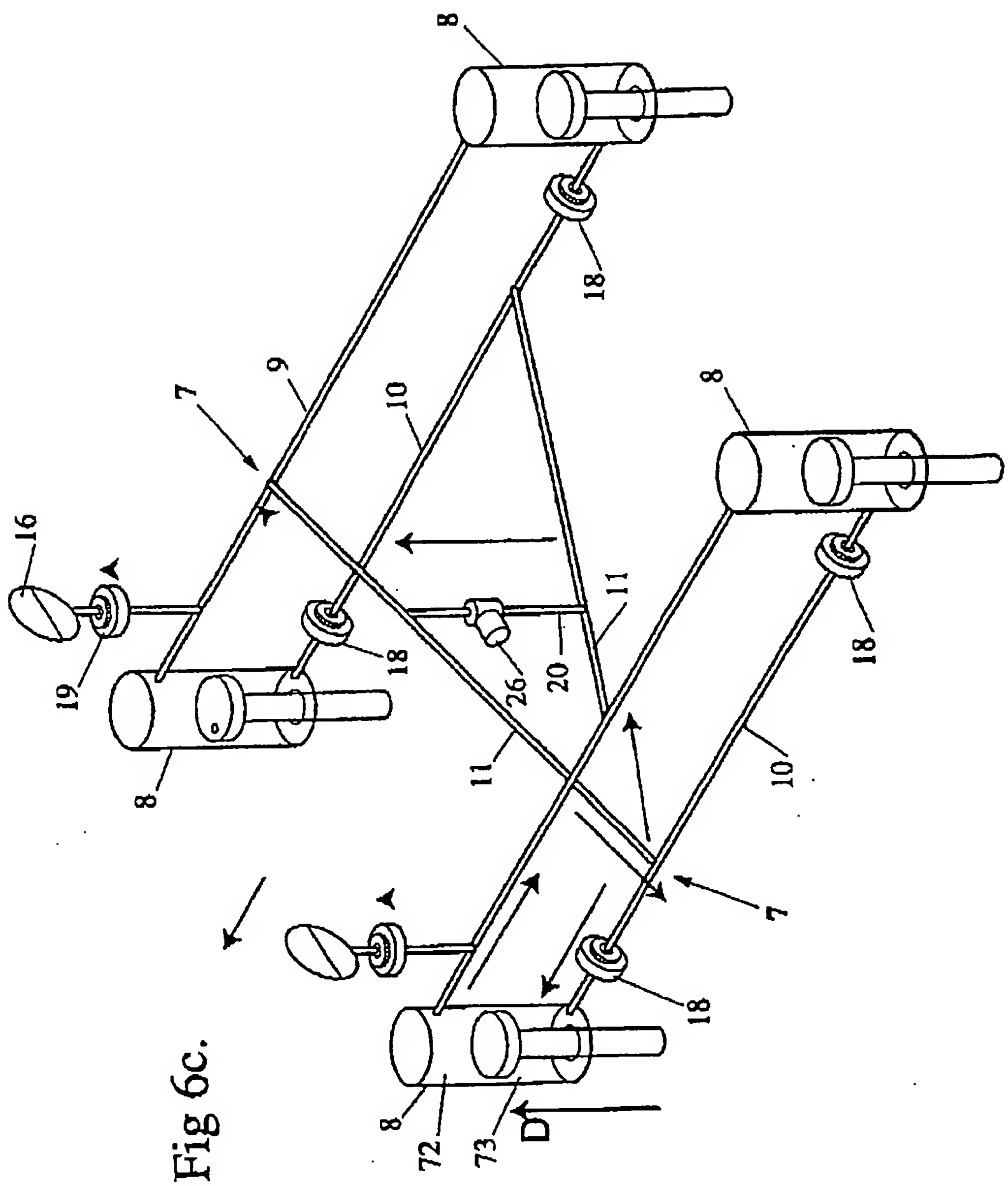


Fig 6c.

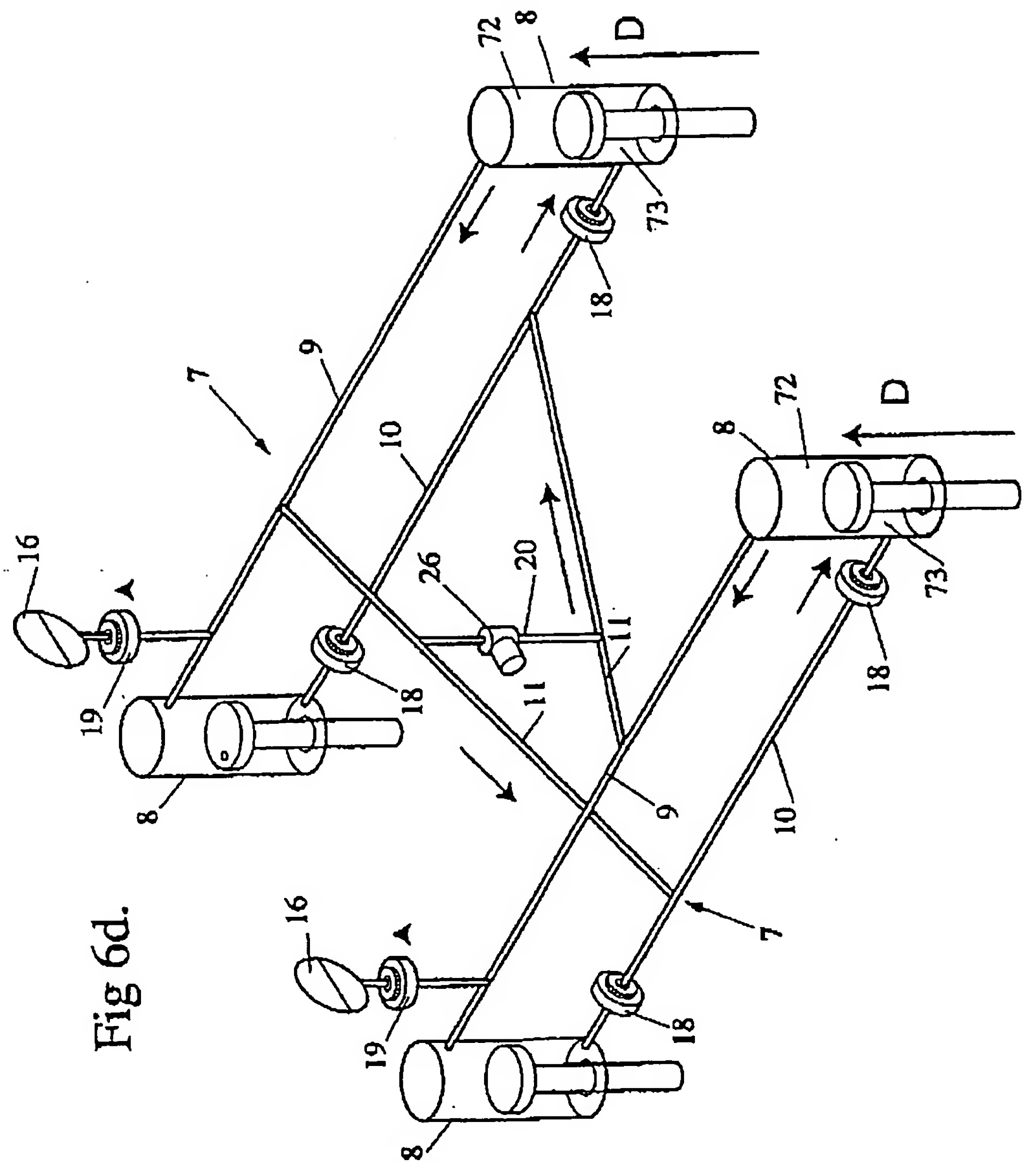


Fig 6d.

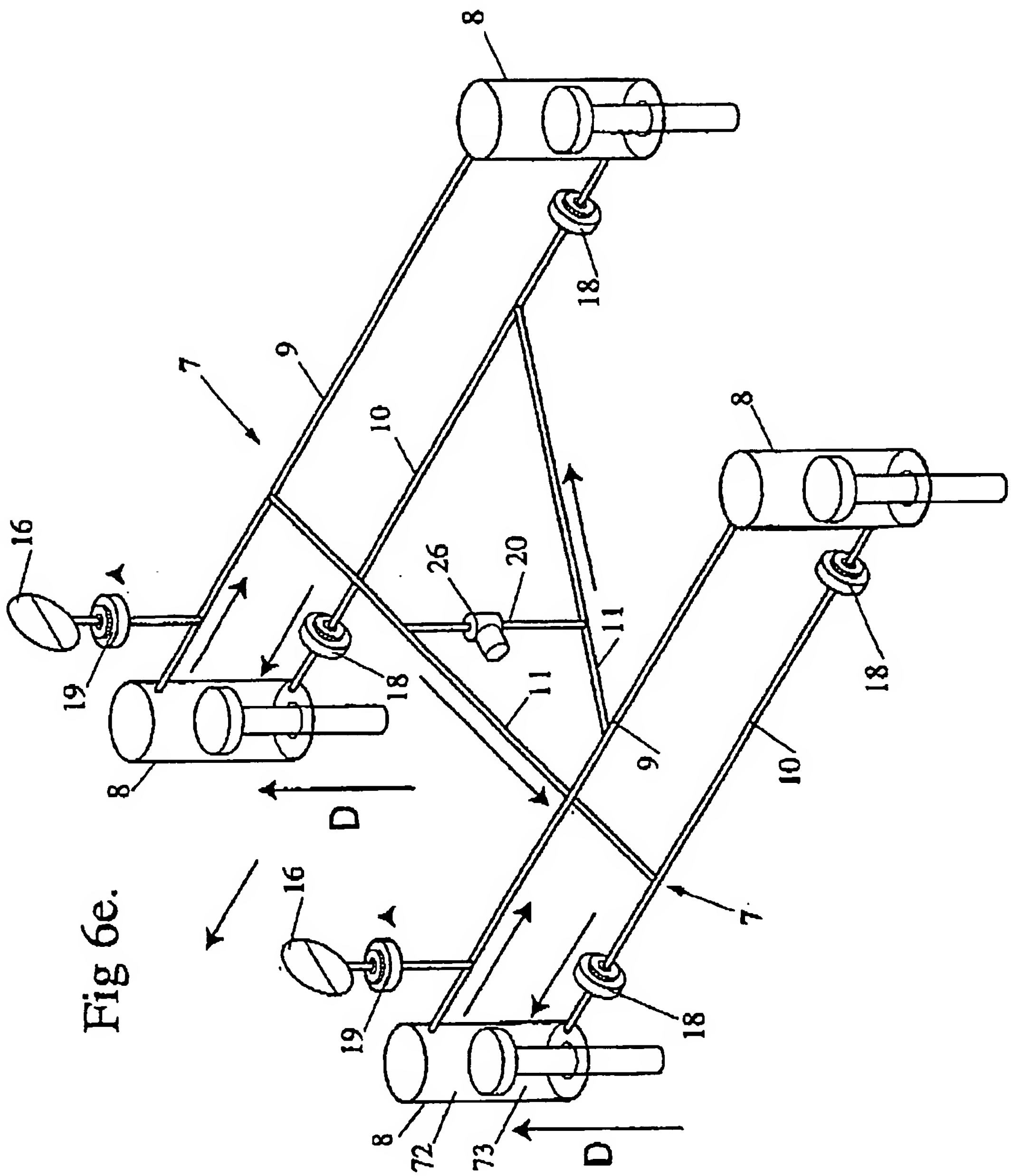


Fig 6e.

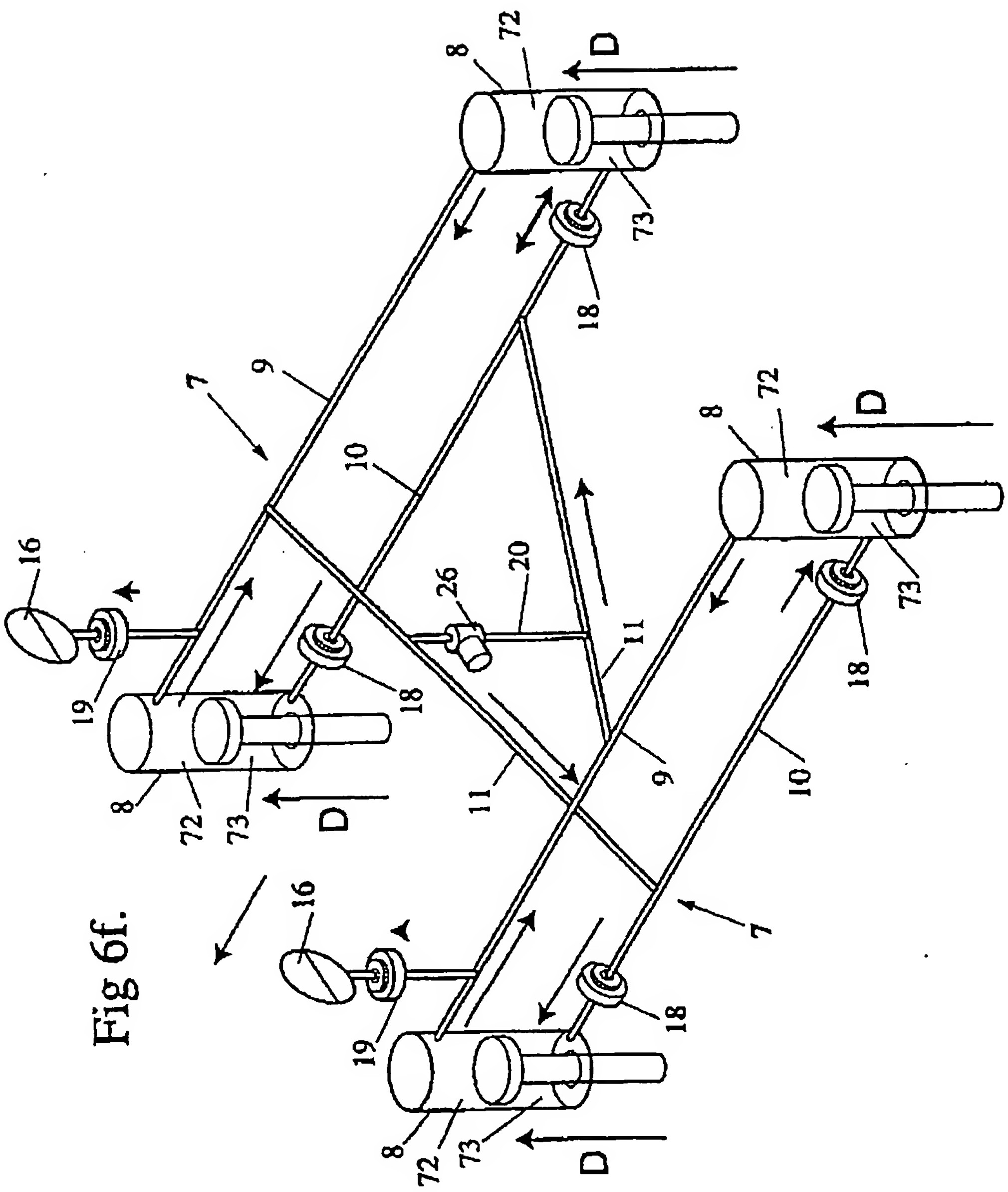


Fig 6f.

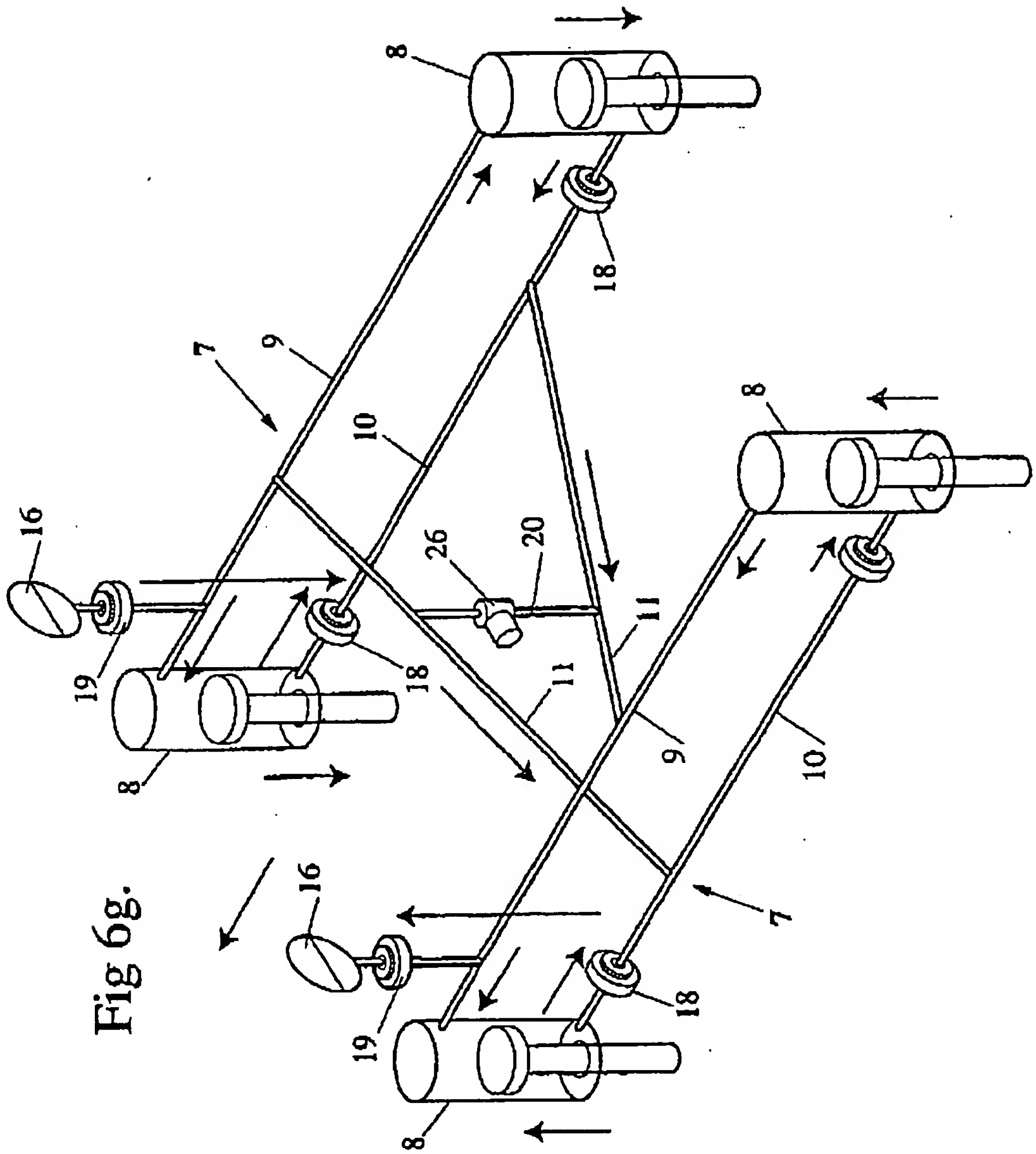


Fig 6g.

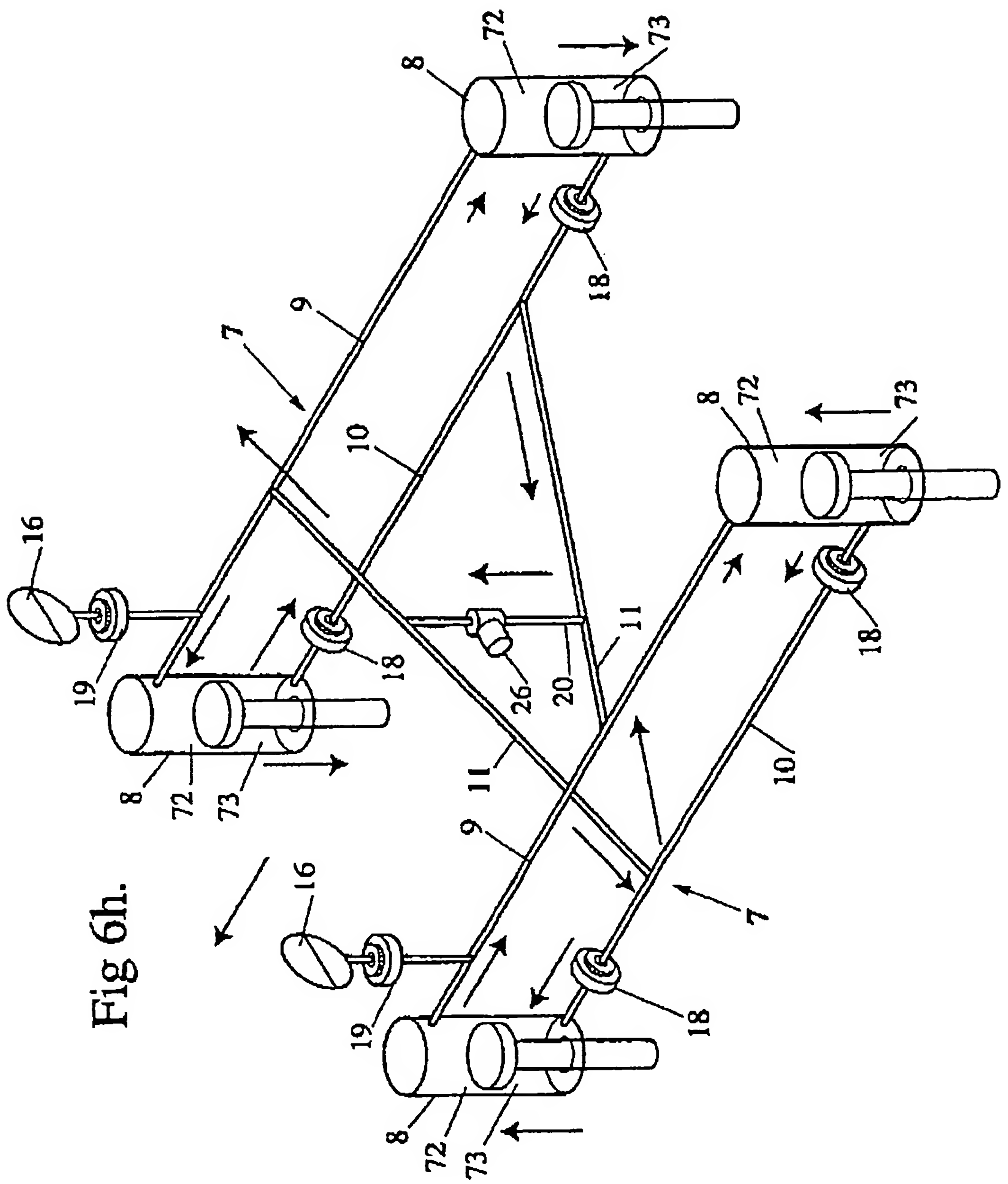


Fig 6h.

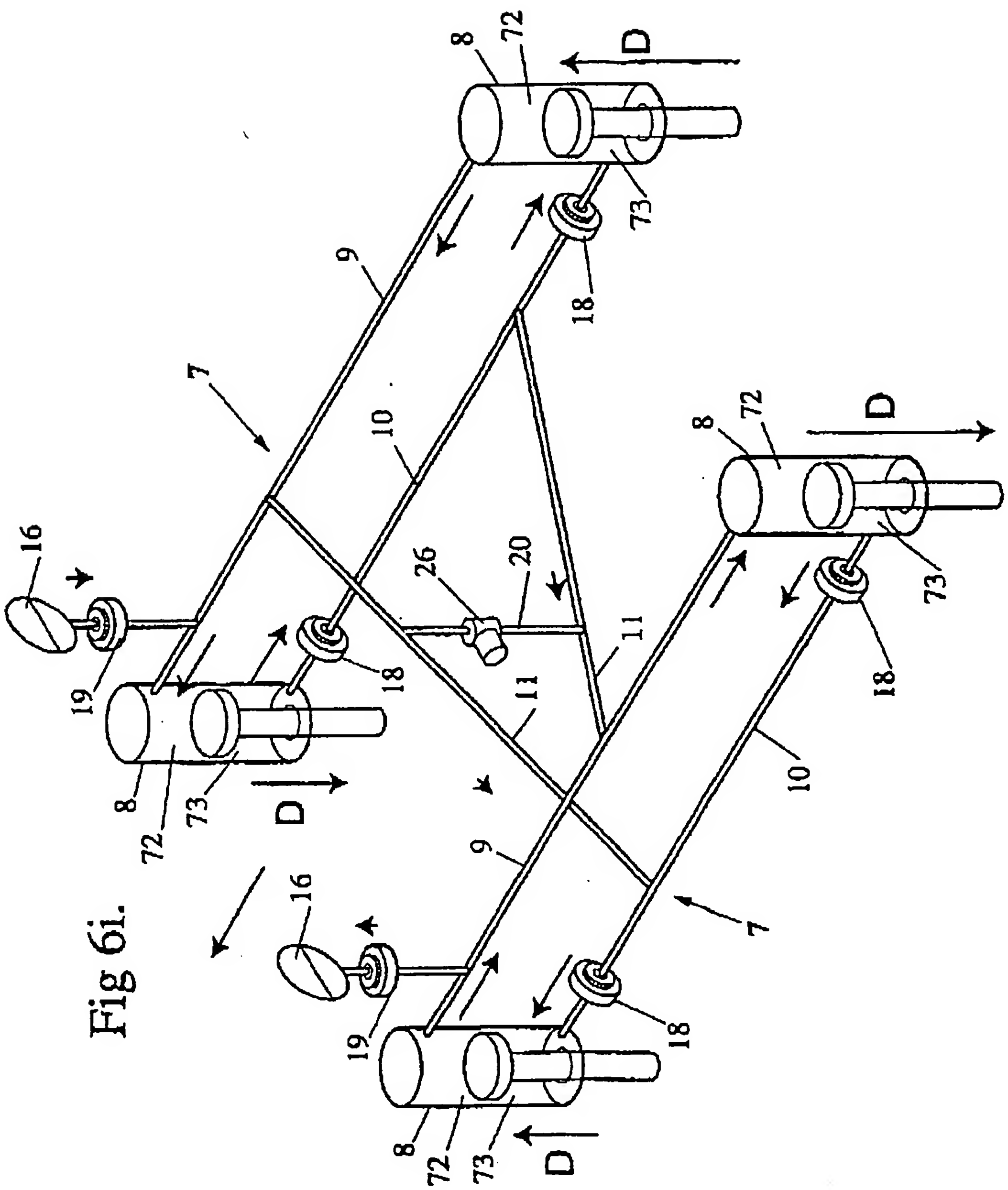

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Fig 6i.

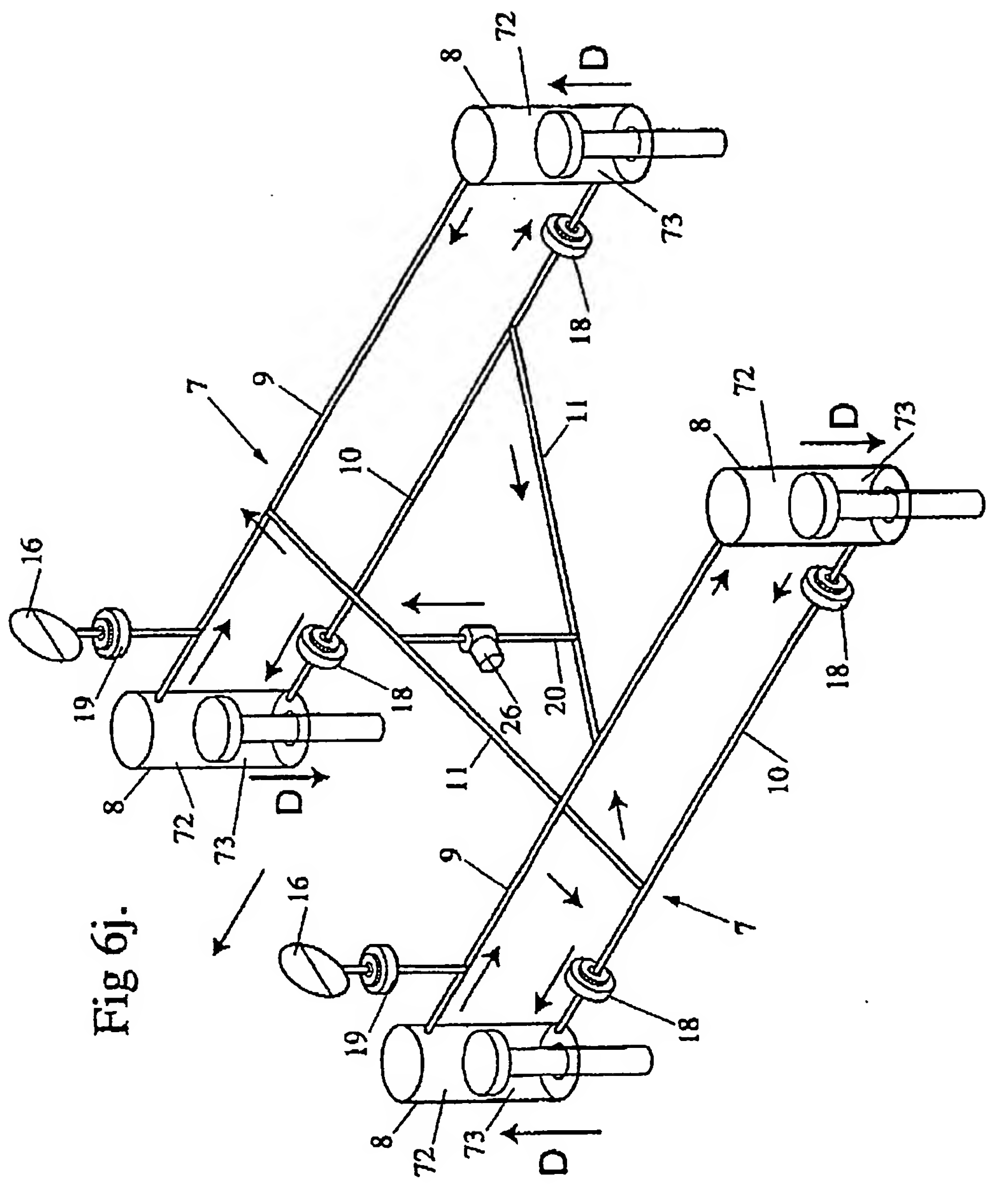


Fig 6j.

Fig 7.

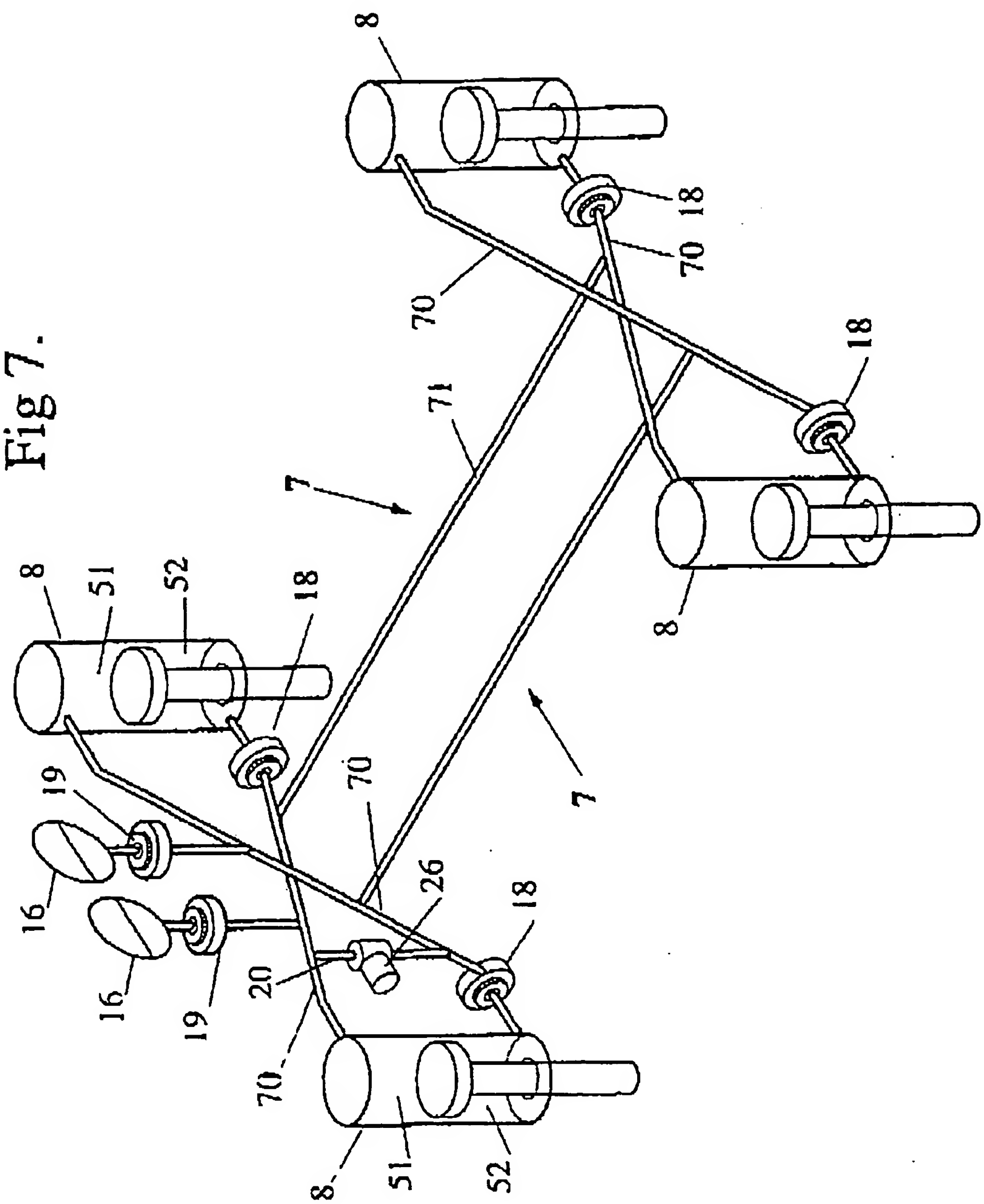


Fig 8.

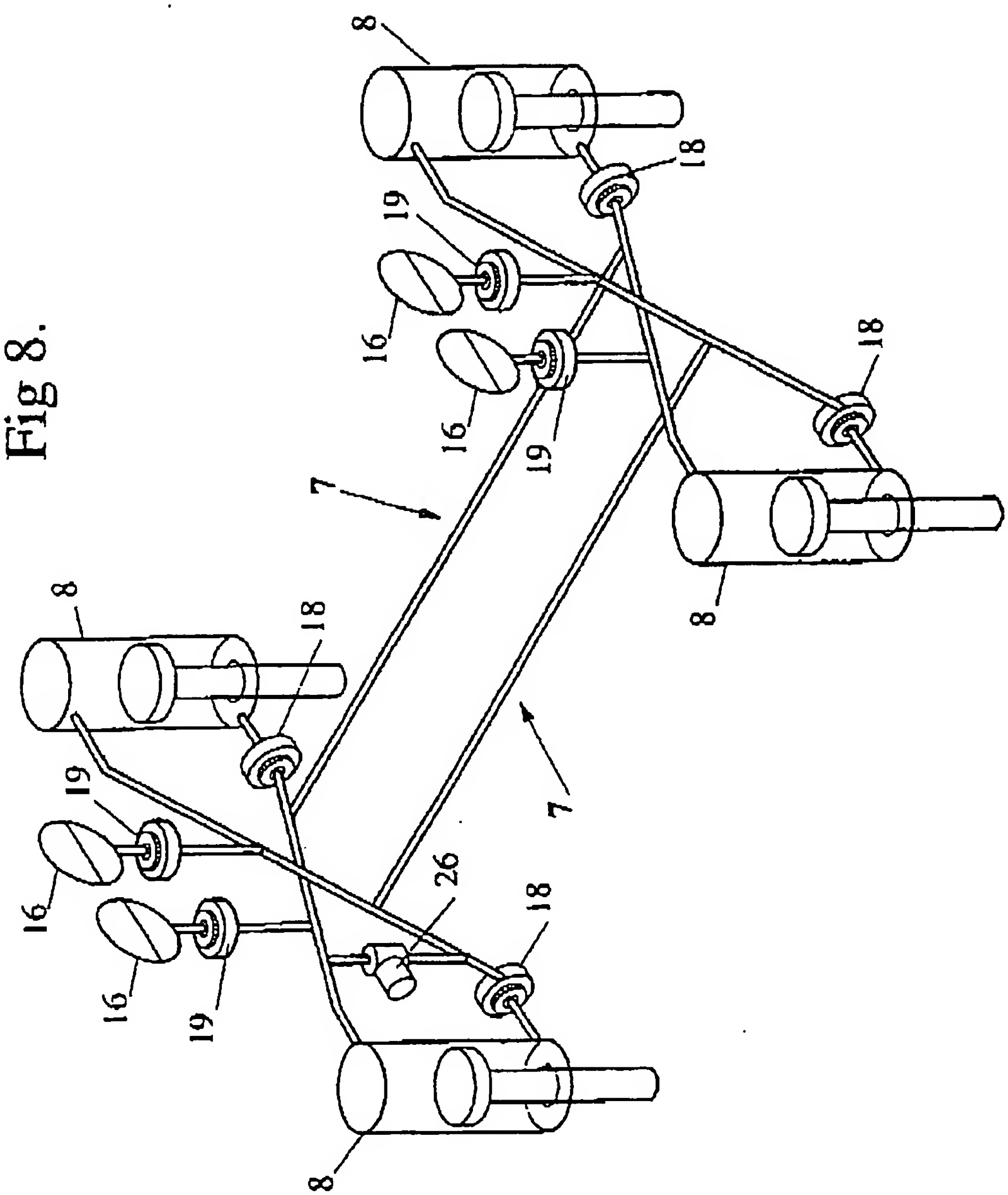
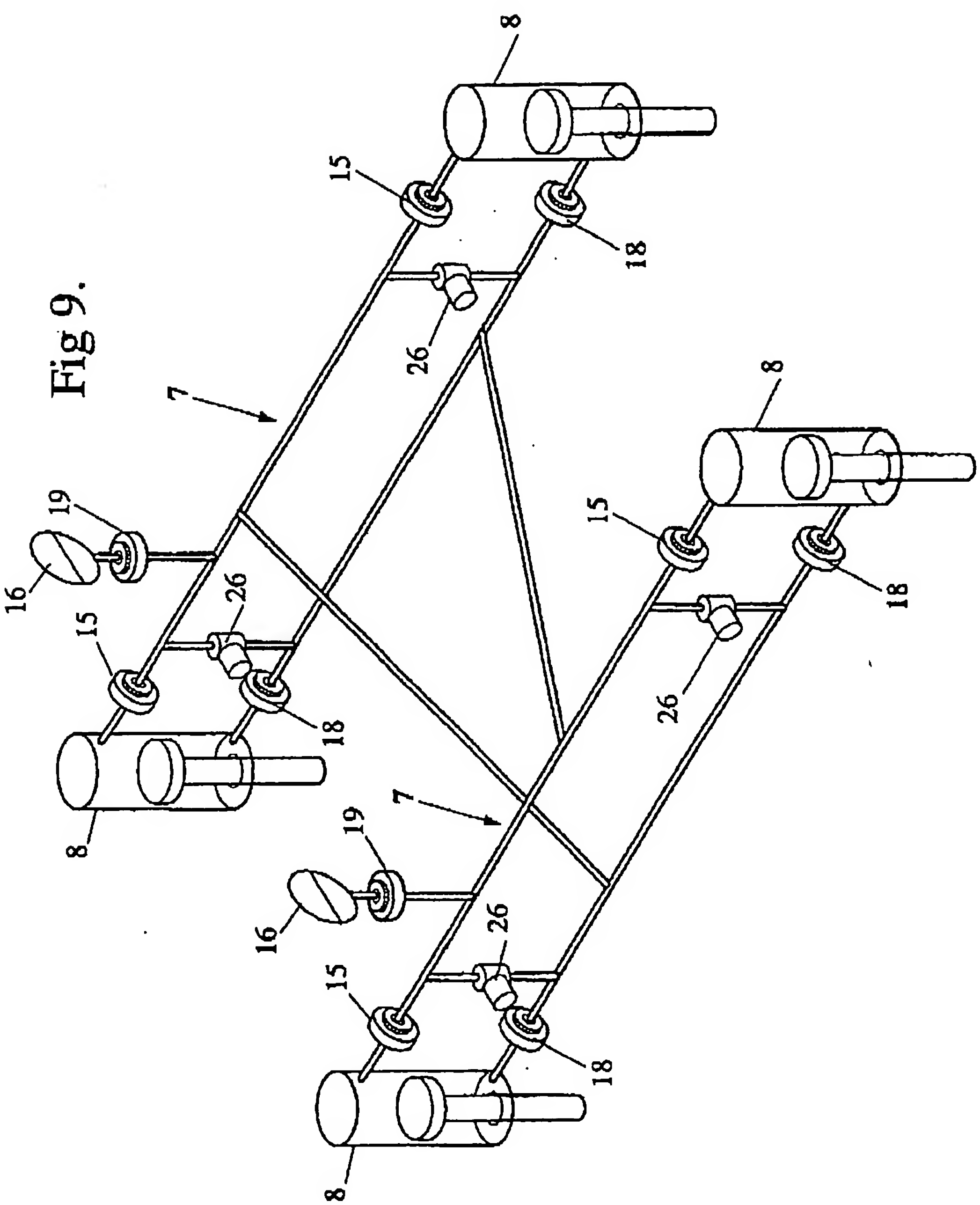


Fig 9.



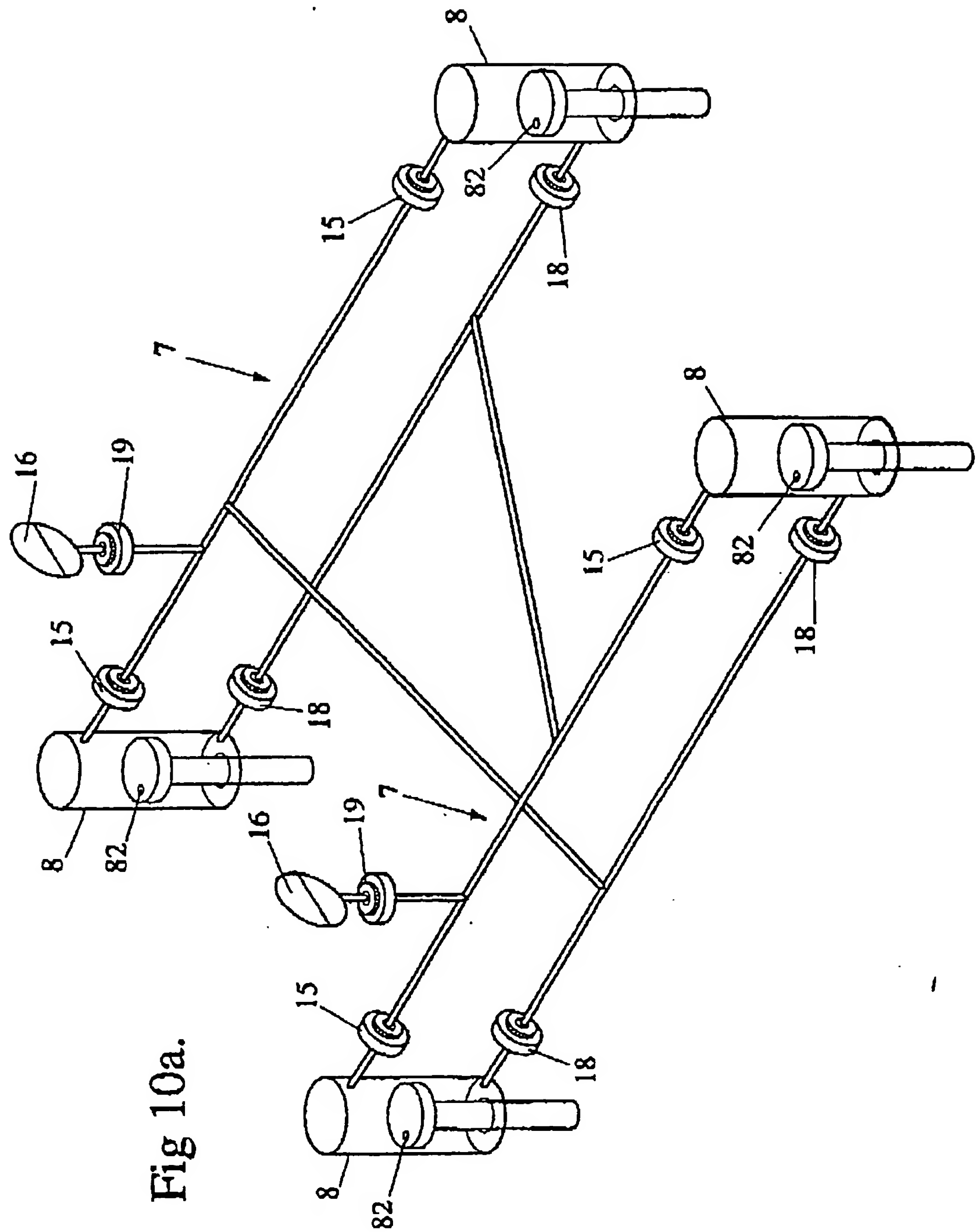


Fig 10a.

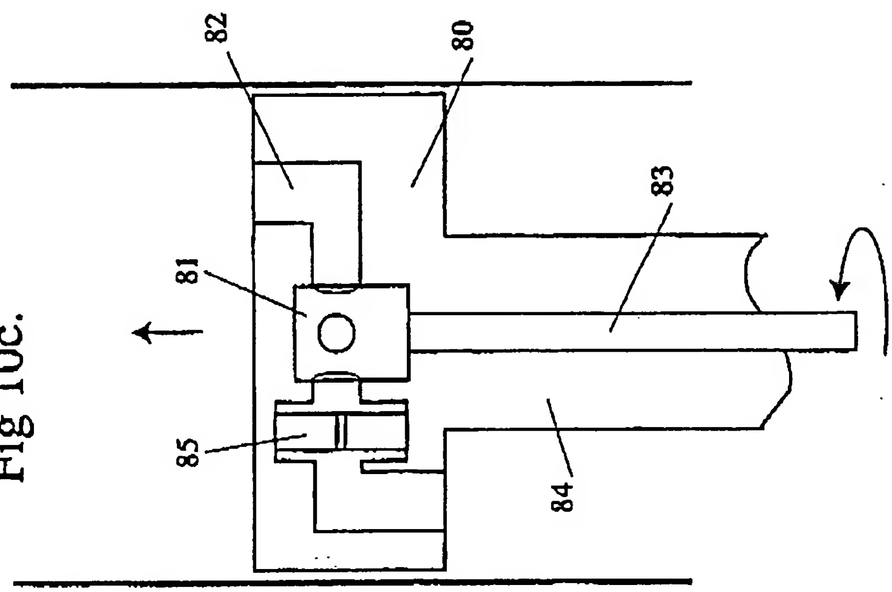


Fig 10c.

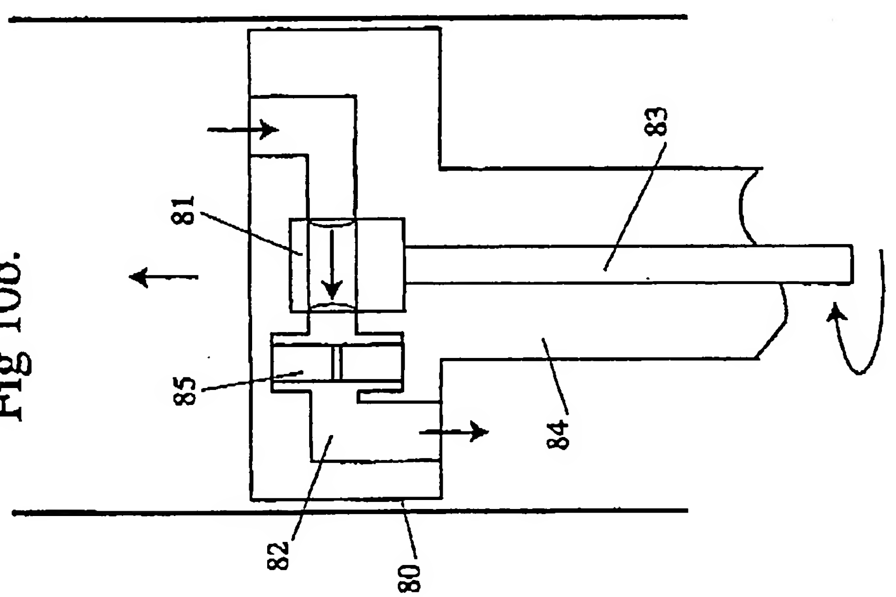


Fig 10b.

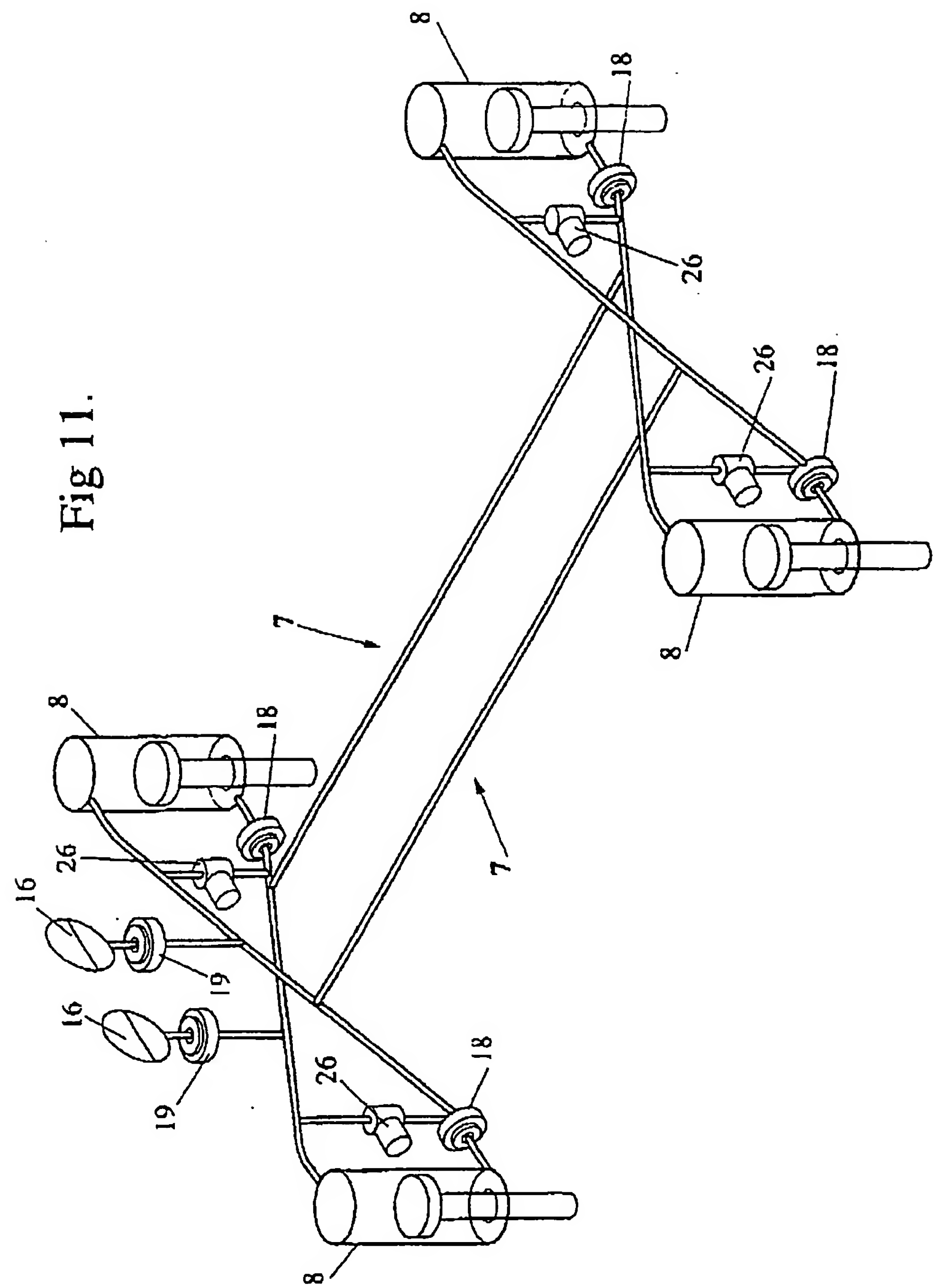
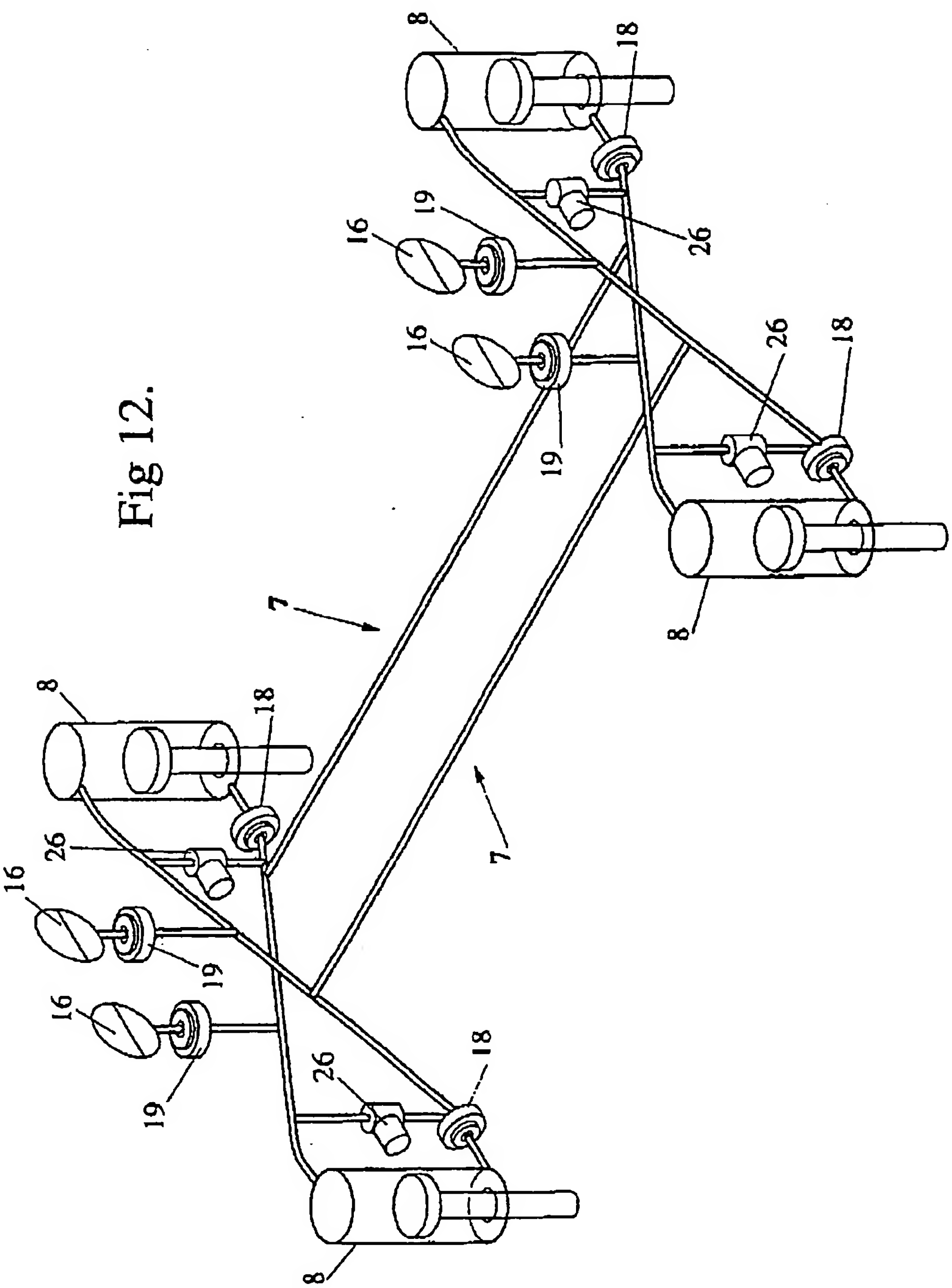


Fig 11.

Fig 12.



PASSIVE RIDE CONTROL FOR A VEHICLE SUSPENSION SYSTEM

This application is the national phase under 35 U.S.C. §371 of PCT International Application No. PCT/AU00/00312 which has an International filing date of Apr. 12, 2000, which designated the United States of America.

FIELD OF THE INVENTION

The present invention is generally directed to vehicle suspension systems, and in particular to vehicle suspension systems incorporating improved passive ride control.

BACKGROUND OF THE INVENTION

The desire for improved ride control in motor vehicles has lead to the development of "active" vehicle suspension systems. Such systems typically use sensors to sense the various ride characteristics of the vehicle, the sensors providing signals to an Electronic Control Unit (ECU). The sensors sense any excessive roll, pitch, four wheel bounce and warp motions of the vehicle and its wheels, and the ECU seeks to actively compensate for this motion by controlling the supply of high pressure fluid from a fluid pump to different actuators acting within the vehicle suspension system, or by controlling the return of high pressure fluid from the actuators to a fluid reservoir. (The warp mode of a suspension system, also known as cross axle articulation, is defined as when one pair of diagonally spaced wheels together move in the opposite vertical direction to the other pair of diagonally spaced wheels with respect to the vehicle body). Active suspension systems which attempt to control all the above-noted ride characteristics are very expensive and complicated and have therefore not proven to be commercially viable. Simpler active systems which only seek to actively control excessive roll motions of the vehicle have therefore also been developed. Similarly, adaptive damping systems are becoming popular as they can be used to influence vehicle motions such as roll, pitch and whole body bounce by changing the damping rates at each wheel without the need for a pump.

All the known active suspension systems however have a number of problems which have prevented commercial acceptance of such systems except in luxury vehicles. The number of components required for such systems have lead to packaging difficulties, with the limited space available for such systems under existing motor vehicles. The complexity of active suspension systems and the high stresses applied to certain components of the system lead to ongoing reliability issues. Furthermore, active systems typically require a large number of components, some of which are specially produced components that can handle high mechanical stresses leading to high manufacturing costs. Also, expensive high pressure and high speed components are typically used in such systems, resulting in relatively higher manufacturing and running costs for active systems when compared with conventional suspension systems. Another disadvantage of active systems is the poor response times generally associated with production feasible versions of such systems. Valves are generally used to control the fluid flow in the system. There is always a certain delay before a valve can be actuated to allow or prevent fluid flow. This delay, together with other delays caused by inadequately defined algorithms controlling the system, can lead to unacceptably poor response times for the active suspension system. Active roll control systems typically respond too slowly when undergoing a quick slalom test for example, the control system being unable to provide adequate control under large changes of inertia.

The Applicant has developed a number of different vehicle system systems which seek to avoid at least some of the problems associated with active suspension systems while providing substantial improvements in the ride of a vehicle. These systems are "passive" and do not require sensors, ECUs or fluid pumps to operate. Such systems are described in Australian Patents 670034, 694762, 671592 and 699388 and International Application No. PCT/AU97/00870, details of which are incorporated herein by reference. These systems do however generally rely on components adapted to handle high pressure fluid.

Adaptive damping systems have been developed specifically to improve the damping function of a vehicle suspension system. These damping systems only require relatively low pressure components when compared with those required in the previously described systems, but provide substantially no roll stiffness. They generally have electrically variable or switchable orifices and preloads which are controlled to provide more appropriate damper forces in a range of predefined conditions to avoid the compromises of a single setting to suit all conditions.

In U.S. Pat. Nos. 5,486,018 and 6,684,496 (Yamaha), there are described interconnected damper systems where the top chamber of at least one pair of laterally or longitudinally adjacent dampers, commonly known as Ôshock absorbers' are connected by a conduit. A number of arrangements are disclosed, providing a range of damping effects. However, none of the arrangements are designed to provide a roll stiffness for the suspension.

In U.S. Pat. No. 4,606,551 (Alfa), there is described an arrangement having dampers, each having an upper and lower chamber. At least one pair of laterally or longitudinally adjacent dampers are connected by conduits respectively connecting the upper chamber of one damper with the lower chamber of the other chamber. A number of damper valves are provided in the connecting conduits to provide various damping effects. No electronic control is required, nor can the arrangement provide a roll stiffness for the suspension.

Although each of the above described adaptive and interconnected damping systems provide an improved damping function over conventional damper arrangements, they do not provide any or only provide minimal control of other ride characteristics of the vehicle. For example, none of the above adaptive or interconnected damping systems provide roll support for the vehicle as they do not have any roll stiffness to enable a degree of roll control for the vehicle, only roll damping. These systems can therefore not be used to provide roll control for the vehicle.

SUMMARY OF THE INVENTION

It is therefore an object of the present invention to provide a damping and roll control system which achieves improved ride control for the vehicle while avoiding at least one of the problems associated with prior art systems.

With this in mind, the present invention provides a damping and roll control system for a vehicle suspension system the vehicle having at least one pair of laterally spaced front wheel assemblies and at least one pair of laterally spaced rear wheel assemblies, each wheel assembly including a wheel and a wheel mounting locating the wheel to permit movement of the wheel in a generally vertical direction relative to a body of the vehicle, and vehicle support means for providing at least substantially a major portion of the support for the vehicle; the roll control system including:

wheel cylinders respectively locatable between each wheel mounting and the body of the vehicle, each wheel cylinder including an inner volume separated into first and second chambers by a piston supported within the wheel cylinder; first and second fluid circuits respectively providing fluid communication between the wheel cylinders by fluid conduits, each said fluid circuit providing fluid communication between the first chambers of the wheel cylinders on one side of the vehicle and the second chambers of the wheel cylinders on the opposite side of the vehicle to thereby provide roll support decoupled from the warp mode of the vehicle suspension system by providing a roll stiffness about a level roll attitude whilst simultaneously providing substantially zero warp stiffness;

each fluid circuit including one or more fluid accumulators for providing roll resilience;

damper means for controlling the rate of fluid flow out of or into at least one chamber of each wheel cylinder, and

selection means for selectively providing fluid communication between the first and second fluid circuits;

the damping and roll control system thereby providing substantially all of the damping of the vehicle suspension system.

The vehicle support means may in certain embodiments of the present invention provide at least substantially all of the support for the vehicle.

The damping and roll control system therefore provides damping for the vehicle suspension and provides a roll stiffness without introducing a corresponding warp stiffness.

Each fluid circuit may in one preferred embodiment include a first fluid conduit providing fluid communication between the first chambers of the wheel cylinders on one side of the vehicle; and a second fluid conduit providing fluid communication between the second chambers of the wheel cylinders on the opposite side of the vehicle; the first and second fluid conduits being in fluid communication.

According to another preferred embodiment, each fluid circuit may include first and second diagonal fluid conduits, each respectively providing fluid communication between the first chamber of one wheel cylinder on one side of the vehicle and the second chamber of the diagonally opposite wheel cylinder on the other side of the vehicle; the first diagonal fluid conduit between one pair of diagonally opposite wheel cylinders being in fluid communication with the second diagonal fluid conduit between the other pair of diagonally opposite wheel cylinders.

According to yet another preferred embodiment, each fluid circuit may include a front fluid conduit providing fluid communication between the wheel cylinders of the front wheel assemblies, and a rear fluid conduit providing fluid communication between the wheel cylinders of the rear wheel assemblies, with the front and rear conduits respectively providing fluid communication between the first chamber of the wheel cylinder at one side of the vehicle with the second chamber of the wheel cylinder at the opposite side of the vehicle, the front and rear conduits being in fluid communication.

It is to be appreciated that other connection arrangements are also envisaged. It is also to be appreciated that the same principles may be applied to vehicles with more than four wheels. For example, to apply the system to a six wheeled vehicle, the additional left hand wheel cylinder will have its first chamber connected to the conduit connecting the first chambers of the other two left hand wheel cylinders, and its second charmer connected to the conduit connecting the second chambers of the other two left hand wheel cylinders.

The connection of the other cylinder to the right hand side of the vehicle similarly communicates first chambers together and second chambers together.

The damper means may be located at or in the wheel cylinders, in the conduits, and/or in a manifold block. The manifold block may be centrally located in the vehicle and may provide the required fluid communication between the first and second conduits to form the first and second fluid circuits, The damper means may be a bi-directional valve (ie. provide controlled flow restriction in both directions), in which case each wheel cylinder requires only one damper valve for one of the first or second chambers. In this case, the associated chamber may try to suck a vacuum if the damper valve is not supplying fluid at the same rate at it is being demanded. This can lead to aeration of the fluid and potential loss of ride control by the system. To avoid this effect, a single direction damper valve may be used to ensure that the wheel cylinder chambers only act through a damper valve when expelling fluid, thereby preventing fluid aeration in the cylinder chambers. Alternatively, the single direction damper valve may be used in parallel with a non-return valve. Alterative, to provide large damping forces with reliable, compact damper valve means, a bi-directional damper means may be provided for each of the first and second chambers of at least one pair of laterally spaced wheel cylinders.

Each said fluid circuit includes at least a first fluid accumulator to allow for changes in the fluid volume of each circuit to thereby provide roll resilience. Also, if a wheel cylinder with differing effective piston areas between the first and second chambers is used (for example a piston having a rod extending from one side only, as in a conventional damper cylinder assembly), the accumulator needs to be able to accommodate the rod volume changes within the system during bounce motions of the suspension. In this case, in roll, the accumulator absorbs a much greater change of fluid volume per unit displacement of the wheel cylinders than it absorbs in bounce as both the effective areas of a first chamber side and a second chamber side are working to displace fluid into the accumulator giving a correspondingly higher stiffness for roll motions of the roll control system than for bounce motions.

Each fluid circuit may include at least one second fluid accumulator to provide increased roll resilience. Between each second accumulator and the respective fluid circuit there may be a roll resilience switching valve. When the vehicle is traveling in a straight line, the valve may be held open to allow the second accumulators to communicate with the associated fluid circuits to provide additional roll resilience, thereby further improving ride comfort. When turning of the vehicle is detected, the roll resilience switching valve is closed to provide a desirable increase in roll stiffness during cornering. The detection of vehicle cornering may be performed in any known manner, using inputs for conditions such as steering rate of change, steering angle, lateral acceleration and vehicle speed. Any or all of these sensors and/or others not cited may be used.

The accumulators may be of the gas or mechanically sprung piston type or the diaphragm type and either or both can be beneficial in increasing the time to maintenance of the system by replenishing fluid lost from the system through leaks past rod seals and out of fittings. Any fluid loss should be minimal, therefore the effect on the operating pressure of the system may be negligible.

At least one of the accumulators in each fluid circuit may have a damper means to control the rate of fluid flow into and/or out of the accumulator. Due to the higher rate of fluid

flow into and out of the accumulators in roll when compared to bounce (as discussed earlier), the effect of the accumulator dampers is greater in roll than in bounce giving a desirable high roll damping to bounce damping ratio. If the accumulators are not damped, the roll damping is determined by the bounce damping, as is the case when using conventional dampers.

Damping the accumulators can also have a detrimental effect to single wheel input harshness as single wheel inputs are also heavily damped by accumulator dampers. To increase comfort in straight line running, it can therefore be advantageous to provide a bypass passage around the accumulator damper valve to permit fluid to bypass the damper for at least one accumulator. The bypass passage includes a valve to open or close the passage. During turning, the valve is in the closed position and the accumulator damper valves are providing high roll damping. In straight line running, the valve is open to reduce the roll and single wheel input damping forces in the system.

The roll control system may have a pressure precharge to allow the accumulators to function and supply fluid in rebound motions of the wheels (where they fall away from the vehicle body). This precharge is preferably about 20 bar for the roll control system with the vehicle at standard unladen ride height.

It may be preferable to use a wheel cylinder design with a rod protruding from one side of the piston through only one chamber. This allows for a simple and cheap cylinder design, but any system precharge pressure acting over the unequal effective piston areas in the first and second chambers produces a not cylinder force. This force may provide some support of the vehicle body although the proportion of vehicle load supported by the roll control system is usually very small and is similar to the degree of support provided by a conventional precharged damper cylinder assembly. The exact amount is determined by the cylinder rod and bore dimensions, system precharge pressure and cylinder to wheel hub lever ratio.

For example, in the case where the first chamber of each wheal cylinder is in compression as the wheels move upwardly with respect to the vehicle body, and the effective area of the piston on the first chamber side is larger than the effective area of said piston on the second chamber side, thereby providing a degree of support of the vehicle body.

If accumulators with a non-linear spring function (ie a hydropneumatic accumulator which has an increasing stiffness in compression and a decreasing stiffness in rebound) are used and the roll control system provides a degree of vehicle support (as outlined above), then as the vehicle rolls due to lateral acceleration, the total volume of fluid in the accumulators can decrease overall, increasing the fluid volume in the roll control system and causing an overall increase in vehicle height (known as "roll jacking"). The degree of vehicle support provided by the roll control system influences the degree of roll jacking.

It may be desirable to produce the inverse of the roll jacking effect such that the average height of the vehicle is lowered during cornering. This effect can be produced in the case where the first chamber of each wheel cylinder is in compression as the wheels move upwardly with respect to the vehicle body, and the effective area of the piston on the second chamber side is larger than the effective area of said piston on the first chamber side, thereby providing a degree of additional load on the vehicle support means, tending to push the vehicle down towards the ground.

Preferably, a simpler arrangement may be used with the cheaper cylinder design which provides vehicle support (discussed above). The resilient means in the first accumulator may include one or more mechanical springs such that the spring rate in the compression direction from the normal static position is lower than the spring rate in the rebound direction from the normal static position, to thereby give the reverse effect of a conventional hydropneumatic accumulator and lower the average height of the vehicle during cornering. Additionally or alternatively, the rebound damping rate of the accumulators may be higher than the compression damping rate to provide a similar vehicle lowering effect and better response to steering inputs during initial cornering (turn-in). Indeed, only rebound damping may be provided for the accumulators, with a non-return valve allowing virtually unrestricted flow in the compression direction.

Ideally, the roll control system should not provide any vertical support of the vehicle. Therefore, in another, alternative preferred arrangement of the present invention, the effective piston areas in the first and second chambers of each cylinder may be similar, the roll control system thereby supporting substantially zero vehicle load. As the amount of vehicle load support provided by the roll control system is one of the main factors controlling the amount of roll jacking inherent in the system, using wheel cylinders with similar effective piston areas in the first and second chambers and which therefore do not provide any vehicle support provides the roll control system with zero roll jacking.

However, in some applications, the use of a cylinder having piston rods extending from both ends thereof can lead to packaging difficulties because of the need to provide clearance for the upwardly extending piston rod. Therefore, according to another preferred arrangement, a piston rod may extend from one side of the piston, the piston rod having as small a diameter as physically possible to minimise the vehicle support provided by the damping and roll control system. In another possible arrangement, a hollow piston rod may extend from one side of the piston, and an inner rod may be supported within the inner volume of the cylinder, the inner rod being at least partially accommodated within the hollow piston rod, the hollow piston rod moving together with the piston relative to the inner rod. This arrangement may be used to minimise the difference in area of the opposing piston faces to minimise the vehicle support provided by the damping and roll control system.

According to an alternative preferred embodiment, the hollow piston rod arrangement of the wheel cylinder may be adapted to also provide a vertical support function for the vehicle. The piston supported in the wheel cylinder may provide an upper and lower chamber. The inner rod when supported within the hollow piston rod defines a rod chamber. This rod chamber may be used as part of a fluid circuit of the roll control system. To this end, the area of the peripheral end of the inner rod may be at least substantially identical to the area of the piston facing the lower chamber. Alternatively, it can be preferable to use a larger lower chamber area than the rod chamber area to induce lowering of the vehicle in roll with increasing roll moment when hydropneumatic accumulators are used in the system.

The upper chamber may be sealed to provide a bounce chamber to provide resilient support for the vehicle. The rod chamber may be vented and, together with the lower chamber, form a respective part of a fluid circuit of the roll control chamber.

It should be noted that the roll moment distribution for the roll control system is determined by the ratio between the effective piston areas of the front wheel cylinders compared to the effective piston areas of the rear wheel cylinders.

Ideally, in most applications, each wheel cylinder should have a constant ratio between the effective piston area on the first chamber side compared to the second chamber side.

One advantage of using cylinders where the piston rod is only provided extending from the one piston face is that the degree of support provided by the cylinders can be varied by varying the support height of the vehicle. As the vehicle is lowered the support provided by the roll control system increases leading to higher roll stiffness. This is an affect of having an increased volume of piston rod introduced into the roll control system.

The support means for at least one pair of laterally spaced wheel assemblies may include first support means which are independent for each wheel assembly, thereby contributing an additional roll stiffness to the suspension system. Both the vehicle support means and the roll control system can together provide the roll stiffness for the vehicle in this arrangement.

Additionally or alternatively, the support means for at least one pair of laterally spaced wheels may include second support means which are interconnected between each wheel thereby contributing substantially zero roll stiffness to the suspension system. This and other vehicle support arrangements that provide little to no roll support and combinations of support arrangements are described in the Applicants' International Application No. PCT/AU97/00870 referred to previously. In such an arrangement, the damping and roll control system can provide substantially all of the roll control for the vehicle. Furthermore if the support means have substantially zero roll stiffness, the damping and roll control system can provide substantially all of the roll control for the vehicle. In this case, neither the support means or roll control system provides significant warp stiffness. This allows for substantially free warp motion of the vehicle wheel assemblies, improving comfort, reactions to single wheel inputs and providing substantially constant wheel loads (and therefore improved traction) in low speed or non-dynamic warp motions when traversing uneven terrain such as in off-road situations.

According to the present invention, the first and second fluid circuits are in fluid communication such that fluid may be transferred therebetween. To this end, at least one bridge passage may interconnect the first and second fluid circuits to provide for said fluid communication. The bridge passage may be provided by a bridge conduit. Alternatively, the bridge passage may be provided within a connector body to which the conduits of the first and second circuits are connected. At least one flow control valve may be provided for controlling the flow through the bridge passage.

One or more accumulators may optionally also be provided for the bridge passage. The flow control valve and accumulator may be provided on a said bridge conduit. According to another possible arrangement, the control valve and/or accumulator may be supported on the connector body. It is also possible for all the damper valves and accumulators previously referred to be located on a common said connector body to simplify the packaging of the system within a vehicle.

The flow control valve may be opened, for example when there is little demand on the roll control system when the vehicle is travelling on a straight road. When the flow control valve is opened, this leads to a "short-circuiting" of the system such that the first and second chambers of each cylinder are allowed to communicate directly. This controlled interconnection of the first and second fluid circuits by the controlled opening of the flow control valve provide a number of operational advantages that lead to improved comfort for the passengers of the vehicle:

a) The damping and roll control system provides no roll stiffness, the only roll stiffness being provided by the vehicle support means.

b) The damping and roll control system no longer effects the roll split of the vehicle, the roll split only being provided by the vehicle support means. If the roll spilt provided by the vehicle support means is between approximately 40 and 60%, this (in combination with the low roll stiffness) acts to reduce vehicle motions leading to "head toss".

c) As there is little resistance to the fluid flow between the chambers of each cylinder except for that provided by the wheel damper valve, the single wheel stiffness is reduced.

d) Because the accumulator damper valves are bypassed, they do not influence the damping function of the damping and roll control system, and the roll damping is the same as the bounce damping.

e) The single wheel damping is (for the same reason) the same as the bounce damping.

f) The bounce damping however remains unchanged when the flow control valve is opened.

The operation of the flow control valve may be controlled by an Electronic Control Unit on the basis of operational parameters such as the lateral acceleration speed and steering rate of the vehicle.

It is also possible for a plurality of bridge passages to be provided interconnecting the first and second fluid circuits. Each bridge passage may be provided with a said flow control valve.

It is also possible that the wheel cylinder include an integral flow control valve and/or damper valve therein. The piston of the wheel cylinder may include a flow control valve and(or damper valve controlling the flow of fluid between the first and second chambers.

The use of a plurality of bridge passages having flow control valves or wheel cylinders having built-in flow control valves facilitates fluid flow between the first and second chambers of the wheel cylinders. This can lead to a reduction in the inertia forces due to fluid flow through the system resulting in improved isolation of high frequency inputs and sharp edge inputs to the vehicle wheels. The effect of inertia forces within the roll control system will be subsequently described in more detail.

As the damping and roll control system can be switched to provide substantially zero roll stiffness, the use of zero roll stiffness support means for all wheels is not viable. However, zero roll stiffness support means may still be used in combination with independent support means providing some roll stiffness. Therefore, the support means for at least one pair of laterally spaced wheels may include first support means for supporting at least a portion of the load on the associated wheel assemblies, said first support means providing independent resilience for each respective wheel and thereby providing a roll stiffness.

Additionally, the support means for at least one pair of laterally spaced wheels may include second support means for supporting at least a portion of the load on the associated wheel assemblies, said second support means providing combined resilience for each associated wheel assembly and thereby providing substantially zero roll stiffness.

It is to be appreciated that the conduit size may be selected to provide a degree of the damping required by the damping and roll control system. Depending on the level of ride comfort required in an application, the conduit size may be selected based on a variety of factors such as fluid inertia,

fluid friction due to viscosity through range of operating temperatures, etc.

The vehicle support means preferably provides most if not all the vertical support for the vehicle. The damping and roll control system however preferably provides little to no vertical support for the vehicle such that the operating fluid pressure within the damping and roll control system can therefore be relatively low when compared with active roll control systems and the Applicants' earlier suspension systems. Theoretically, if the roll control system provides no vertical support for the vehicle, the operating pressure may be only atmospheric pressure, the system has no precharge pressure.

The damping and roll control system of the suspension system according to the present invention can therefore use low pressure components. The wheel cylinders can be constructed using standard vehicle damper and sealing technology. This leads to substantial manufacturing cost savings when compared to higher pressure systems. Also, comfort and NVH problems associated with higher pressure systems such as "stiction" between components are minimised in low pressure systems, the stiction levels being similar to that present in a conventional damper cylinder assembly.

Such a damping and roll control system can be installed in existing vehicle suspension systems, the dampers used in such systems being replaced or adapted for use as the wheel cylinders of the roll control system according to the present invention. The existing vehicle support means supporting the vehicle such as conventional steel or pneumatic springs can be retained. Alternatively, the vehicle support means may be replaced by support means that provide little to no roll support as described previously. This is possible because the damping and roll control system also provides a roll stiffness for the vehicle suspension system.

According to a further aspect of the present invention, there is provided a method of controlling the roll damping and roll stiffness of a damping and roll control system for a vehicle suspension system, the damping and roll control system including:

- wheel cylinders respectively locatable at wheel assemblies of the vehicle, each wheel cylinder including an inner volume separated into first and second chambers by a piston supported within the wheel cylinder; and first and second fluid circuits respectively providing fluid communication between the wheel cylinders by fluid conduits, each said fluid circuit providing fluid communication between the first chambers of the wheel cylinders on one side of the vehicle and the second chambers of the wheel cylinders on the opposite side of the vehicle to thereby provide roll support decoupled from the warp mode of the vehicle suspension system by providing a roll stiffness about a level roll attitude whilst simultaneously providing substantially zero warp stiffness;
- damper means for controlling the rate of fluid flow into and out of at least one chamber of each wheel cylinder.
- the method including opening the selection means to provide fluid communication between the first and second fluid circuits when the lamping and roll system is required to provide a relatively low level of roll stiffness and roll damping; and
- closing the selection means to prevent fluid communication between the first and second fluid circuits when the damping and roll system is required to provide a relatively high level of roll stiffness and roll damping.

The fluid flow may be bypassed from at least a substantial portion of the fluid conduits by opening the selection means when there is a single wheel input or two wheel parallel bump input to the damping and roll control system. The line damping and fluid inertia effects on the damping of the control system can therefore be minimised at such wheel inputs.

It is also envisaged that the entire fluid flow be bypassed from the fluid conduits at the predetermined wheel inputs. This can for example be achieved by providing a control valve within the wheel cylinder as hereinbefore described.

Damping means such as single and bi-directional damper valves may be provided through which the bypassed fluid flow passes, these damping means clearly controlling the damping of the control system during this operational mode.

It will be convenient to further describe the present invention with respect to the accompanying drawings which illustrate preferred embodiment of the invention. Other embodiments of the invention are possible, and consequently the particularity of the accompanying drawings is not to be understood as superseding the generality of the preceding description of the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a partially schematic view of a first preferred embodiment of a roll control system according to the present invention mounted on wheel assemblies of a vehicle;

FIG. 2 is a schematic view of second preferred embodiment of a roll control system according to the present invention;

FIG. 3 is a detailed view of a preferred embodiment of a wheel cylinder and wheel damper valve arrangement according to the present invention;

FIG. 4 is a schematic view of another preferred embodiment of a wheel cylinder and wheel damper valve according to the present invention;

FIG. 5 is a schematic view of yet another preferred embodiment of a wheel cylinder according to the present invention;

FIGS. 6*a* to 6*j* are schematic views showing the fluid flow within the damping and roll control system according to the present invention under different wheel inputs to the vehicle;

FIG. 7 is a schematic view of a third possible arrangement of a roll control system awarding to the present invention;

FIG. 8 is a schematic view of a fourth possible arrangement of a roll control system according to the present invention;

FIG. 9 is a schematic view of a fifth possible arrangement of a roll control system according to the present invention;

FIG. 10*a* is a schematic view of a sixth possible arrangement of a roll control system according to the present invention;

FIGS. 10*b* and 10*c* is a schematic cross-sectional view of the wheel cylinder piston with an internal flow control valve and damper valve for the arrangement shown in FIG. 10*a*;

FIG. 11 is a schematic view of a seventh possible arrangement of a roll control system according to the present invention; and

FIG. 12 is a schematic view of an eighth possible arrangement of a roll control system according to the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring initially to FIG. 1, there is shown the front wheel assemblies 2 and rear wheel assemblies 3 of a vehicle,

the body of the vehicle not being shown for clarity reasons. Each front wheel assembly 2 includes a wheel mounting 5 in the form of a wish-bone link contributing to the location of a respective wheel 4 (a second wishbone may be used but is omitted for clarity, other types of wheel locating links may be used). The rear wheel assemblies 3 have a common solid axle 6 to which each wheel 4 is mounted. The vehicle support means 17*a*, 17*b* for supporting the vehicle are shown fixed to the front wishbones 5 and adjacent the rear wheel axle 6 and include independent torsion bars 22 and a pair of air springs 23 interconnected by a conduit 21. The independent form of front vehicle support means 17*a* shown as torsion bars provide a roll stiffness and the interconnected form of rear vehicle support means provides practically no roll stiffness because fluid is allowed to flow between the air springs 23 via the conduit 21. Alternative vehicle support means can also be used, such as any known independent support means or low roll stiffness support means, or any combination different support means. For example, the vehicle may be supported entirely by independent coil springs. Alternatively, it may be supported by a combination of independent coil springs and interconnected air springs at one or both ends of the vehicle. Any combination of independent, combined or zero roll stiffness support means may be used on the front and rear of the vehicle. Many variations are shown and described in the Applicants International Application No. PCT/AU97/00870 and incorporated herein by reference.

A damping and roll control system 1 interconnects the front and rear wheel assemblies 2, 3 and includes a wheel cylinder B respectively provided for each front wheel assembly 2 and rear wheel assembly 3, and a pair of fluid circuits 7,

The configuration of the damping and roll control system 1 can be more readily understood by referring to FIG. 2. (Alterative possible arrangements of the damping and roll control system 1 are discussed later and shown in FIG. 5 onwards) of this arrangement and of subsequent arrangements. It should be noted that corresponding features are designated with the same reference numeral for clarity reasons. Each wheel cylinder 8 has an inner volume 60 separated into an upper chamber 51 and a lower chamber 52 by a piston 53. Piston rods 54, 55 extend from both sides of the piston 53 in the wheel cylinder 8 shown in FIGS. 2 and 3. Each fluid circuit 7 further includes an upper conduit 9 connecting the upper chambers 51 of one pair of longitudinally adjacent wheel cylinders 8, and a lower conduit 10 interconnecting the lower chambers 52 of the opposing pair of longitudinally adjacent wheel cylinders 8. As best shown in FIG. 1, each fluid circuit 7 may further include a cross conduit 11 which connects the lower conduit 10 with the upper conduit 9. The two cross conduits 11 are themselves connected by a bridge passage 20.

Wheel damper valves 18 can be provided on the lower conduit 10, a respective wheel damper valve 18 being provided for the lower chamber 52 of each wheel cylinder 8. Wheel damper valves 15 can also be provided on the upper conduit 9, a respective upper wheel damper valve 15 being provided for each upper chamber 51 of each wheel cylinder 8.

An accumulator 16 can also be provided for each fluid circuit 7. In the arrangement shown in FIGS. 1 and 2, each accumulator 16 is provided at the junction between the lower conduit 10 and cross conduit 11. An accumulator damper valve 19 is provided at the mouth of each accumulator 16.

A flow control valve 26 is provided on the bridge passage 20 for controlling the flow of fluid through the bridge passage 20. The flow control valve 26 is controlled by an electronic control unit (ECU) 27 which controls the valve 26 as a function of different operational parameters. FIG. 2 shows the ECU 27 receiving signals from a steering input sensor 35 located on a steering wheel 40 of the vehicle, a lateral acceleration sensor 36 and a speed sensor 37.

As the wheel cylinders 8 shown in FIG. 2 include piston rods 54, 55 extending from both sides of the piston 53 such a wheel cylinder 8 provides no support for the vehicle. The support is therefore substantially entirely provided by the vehicle support means 17*a*, 17*b* which are schematically shown as coil springs in FIG. 2.

FIG. 3 is a detailed view of the wheel cylinder 8 of FIG. 2 and its associated wheel damper valves 15, 18. The lower wheel damper valve 18, which is schematically shown in FIG. 3, provides a restriction of fluid flow to the lower chamber 52 while allowing relatively unimpeded flow of fluid from that lower chamber 52. By comparison, the upper damper valve 15, also shown schematically in FIG. 3, restricts the flow of fluid from the upper chamber 51 while at the same time providing relatively unimpeding flow of fluid to the upper chamber 51. This arrangement allows a positive pressure to be maintained in the upper and lower chambers 51, 52 and the upper and lower conduits 9, 10 to thereby prevent a vacuum being formed therein. This which can result in aeration of the fluid which can cause the damping and roll control system 1 to not operate properly, part of a "gimbal" style mount for this "through rod" cylinder design is shown at 49.

FIG. 4 shows an alternative possible arrangement of the wheel cylinder 8 according to the present invention. This wheel cylinder 8 includes a "dummy" rod 61 extending internally through the inner volume 50 of the wheel cylinder 8. The dummy rod 61 is slidably accommodated within a hollow rod 62 which is itself supported on the piston 60. The piston 60 and hollow rod 62 which can therefore slide over the dummy rod 60. This arrangement minimises the difference in area between the upper face 60*a* and the lower face 60*b* of the piston 60. The wheel cylinder 8 according to this arrangement will therefore provide minimal support for the vehicle.

The wheel cylinder shown in FIG. 4 could also be adapted to provide a support function for the vehicle as well as provide for roil control as shown in FIG. 5. The dummy rod 61 when located within the hollow rod 62 defines a rod chamber 83. The dummy rod 61 has an area 61*a* at its peripheral end. The diameter of the dummy rod 82, and therefore the end area 61*a* may be sized such the area of the lower face 60*b* of the piston is at least substantially the same as the end area 61*a* of the dummy rod. By sealing the upper chamber 51 and venting the rod chamber 63 along a vent passage 64 provided through the dummy rod 61 so that it becomes part of the roll control system, this allows the wheel cylinder to also function as a support for the vehicle. The sealed upper chamber 51 will in this configuration act as a bounce chamber to provide resilient support for the vehicle such that the need for other support means such as coil springs can be eliminated. The lower chamber 52 and rod chamber 63 can then respectively form part of the fluid circuit of the roil control system.

FIGS. 6*a* to 6*j* schematically shows the fluid flows through the damping and roll control system 1 during different wheel inputs and vehicle motions. The arrow designated with the letter D represents the magnitude and direction of the wheel input into the wheel cylinder 8 immediately adjacent the arrow. The remaining arrows rep-

resent the direction and magnitude of the fluid flows within the damping and roll control system. In all of the following Figures, the front of the vehicle is located at the top left hand corner of each Figure.

Single Wheel Input

FIGS. *6a* to *6c* shows the fluid flows in response to a single wheel input. It should be noted that the wheel cylinders **8** are shown having a piston **70** with a single piston rod **71** extending from the bottom face of the piston **70**. Such a wheel cylinder **8** provides a small degree of support for the vehicle due to the difference in the areas of the upper and lower piston faces of the piston **70**. The degree of support provided by the wheel cylinder **8** can however be minimised by having the diameter of the piston rod **71** as narrow as physically possible.

FIGS. *6a* and *6b* show the fluid flow when the flow control valve **26** in the bridge passage **20** is closed. In FIG. *5a*, a wheel input D is provided to the left rear wheel cylinder **8**. This results in an upward movement of the piston **70** therein which reduces the volume of the upper chamber **72** of that wheel cylinder **8**. Because the fluid is incompressible, some fluid is transferred along the upper conduit **9** to the accumulator **16**. Because of the increase in volume in the lower chamber **52** of the rear left wheel cylinder **8**, fluid must be drawn from another part of the damping and roll control system **1**. To this end, fluid can be drawn from the accumulator **16** located on the top conduit **9** on the right hand side of the vehicle, through the cross conduit **11** to the lower conduit **10** on the left hand side of the vehicle. No fluid is therefore drawn from of directed to the other wheel cylinders **8** and there is therefore no displacement of the piston rod **71** of the other wheel cylinders **8**. It should be noted that the lower wheel damper valve **18** associated with the left rear wheel cylinder **8** and the accumulator damper valves **19** control damping of the vehicle motion.

FIG. *6b* shows the effect of a single wheel input D into the left front wheel cylinder **8**. In comparison with FIG. *6a*, a greater magnitude of fluid flow occurs within the damping and roll control system **1**, the fluid forced from the upper chamber **72** of the left front cylinder **8** being directed to the accumulator **16** on the left hand side of the vehicle, with further fluid being drawn from the accumulator **16** of the right hand side of the vehicle to the lower chamber **73** of the left front cylinder **8**. There is again no displacement of the piston rod **71** of the remaining wheel cylinders **8**. In this situation, the magnitude of flow to and from the accumulators are significantly higher than when the single wheel input is to one of the rear wheel cylinders **8**. The damping of the vehicle motion is therefore largely controlled by the accumulators.

In FIG. *6c*, the fluid flow valve **26** is open allowing flow through the bridge passage **20**. This valve **26** is opened when the vehicle is not undergoing any motion that would place a demand on the damping and roll control system **1**. The same wheel input D into the front left wheel cylinder **8** simply results in fluid being delivered from the upper chamber **72** thereof along the upper conduit **9**, through the cross conduit **11**, the bridge passage **20**, the other cross conduit **11**, the lower conduit **10**, back to the lower chamber **73** of the left front wheel cylinder **8**. The fluid in other words flows from the upper chamber **72** to the lower chamber **73** of the wheel cylinder **8** with little to no fluid flow to and from the accumulator **16** on each fluid circuit **7**. The damping is therefore entirely controlled by the lower wheel damper valve **18** associated with the left front wheel cylinder **8**. The single wheel damping in this situation is therefore the same as the bounce damping of the system.

Two Wheel Bounce

FIGS. *6d* and *6e* show the fluid flows in the damping and roll control system **1** when two wheel bounce is experienced. In both figures, the flow control valve **26** remains closed. FIG. *6d* shows a wheel input D being applied to the two rear wheel cylinders **8**. The reduction in the volume of the upper chamber **72** of each of the rear wheel cylinders **8** results in fluid being pushed through the top conduits **9** along the cross conduits **11** to the lower chambers **73** of the adjacent rear wheel cylinder **8**. There is no fluid flow to or from the front wheel cylinders **8** or the accumulators **19** and the damping is controlled by the lower wheel damper valves **18** of each of the said cylinders **8**.

In FIG. *6e*, there is shown a wheel input D to the two front wheel cylinders **8**. This results in a corresponding fluid flow of fluid from the upper chamber **72** of the front wheel cylinder **8** to the lower chamber **73** of the adjacent front wheel cylinder **8**. The damping is again controlled by the lower wheel damping valves **18**, with little to no fluid flow to the accumulators **16**.

Four Wheel Bounce

FIG. *6f* shows the fluid flow in the right control system **1** when a wheel input D is provided to all four wheel cylinders **8**, with the flow control valve **26** remaining closed. The fluid displaced from the upper chambers **72** of the wheel cylinders **8** on one side of the vehicle is displaced through the cross conduit **11** and the lower conduit **10** to the lower chambers **73** of the wheel cylinders **8** of the opposing side of the vehicle. There is little to no flow to and from the accumulators **16** and the damping is controlled by the lower wheel damper valves **18**.

Roll

FIGS. *6g* and *6h* show the fluid flow control valve **26** is closed in FIG. *6g* and is opened in FIG. *6h*. The roll motion of the vehicle results in a wheel input D being provided to the wheel cylinders **8** on the left hand side of the vehicle in an upward direction, the wheel input to the wheel cylinders **8** of the right hand side of the vehicle being in a downward direction. The next result of the fluid flow is that a substantial amount of fluid must be drawn from the accumulator **16** of one fluid circuit **7**, while the accumulator of the other fluid circuit **7** must accommodate a substantial amount of fluid. The accumulators **16** and their associated damper valves **19** therefore have a substantial effect of the damping and roll stiffness of the roll control system **1** when the flow control valve **26** is closed.

By comparison, in FIG. *6h*, because the flow control valve **26** is open, the fluid flow is "short circuited" such that fluid is simply transferred between the upper and lower chambers **72**, **73** of each wheel cylinder **8** with little to no fluid being drawn or supplied to each of the accumulators **16**. In this arrangement, the accumulator **16** have no influence of the roll stiffness of the damping and roll control system **1**.

Articulation

FIGS. *6i* and *6j* shows the fluid flows within the damping and roll control system **1** during articulation motion of the vehicle wheels. FIG. *6i* shows the fluid flows when the flow control valve **26** is closed, FIG. *6j* showing the fluid flow with the fluid control valve **26** open.

Referring to FIG. *6i*, the wheel input D due to the articulation motion of the wheels simply result in the transfer of fluid between the upper chambers **72** and the lower chambers **73** of each pair of wheel cylinders **8** in each fluid circuit **7** with no transfer of fluid between the fluid circuits **7**, by comparison, in FIG. *6j*, the opening of the flow control valve **26** again results in "short circulating" of the fluid flow such that there is simply a transfer of fluid between the upper and lower chambers **72**, **73** of each wheel cylinder **8**.

Any suspension system which includes an arrangement of interconnected fluid cylinders (such as the present invention) responds to inputs by producing forces which can be placed into four categories. The first is spring forces produced by compression or wind up of the fluid and/or mechanical springs in the system (and other sources of resilience such as hose expansion), this spring force being a function of the displacement of one or more of the fluid cylinders. The effect of the spring force is most noticeable at low frequencies.

The second category of forces is static friction forces which occur when wheel cylinder motion is initiated, or when the direction of motion is reversed. These state friction forces are often referred to as "stiction" forcas or "breakout friction" forces and are due to the friction between the rod and piston seals and the respective rod and bore surfaces.

The third category of forces is damping forces which are a function of velocity. Primarily these damping forces are regulated by orifices, shims and springs in the damper valves **15**, **18**, **19**. A component of the total system damping is generally provided by "line damping", ie. the flow of fluid along the conduits interconnecting the wheel cylinders in the system. The cross sectional areas of the wheel cylinders and the fluid conduits and the lenghts of the fluid conduits should be designed to ensure that the level of line damping provided is of an acceptably low level for the different flows possible due to the motions of the suspension in the modes discussed above.

The fourth category of forces is the inertia forces, due primarily to the acceleration of the fluid through the system. Therefore, the inertia effect is most noticeable at high frequencies and may provide reduced isolation of high frequency inputs and sharp edge inputs resulting in body vibration and noise. Consider a theoretical system consisting of a cylinder with a piston area Ap connected to a line of length L and area Al and an incompressible fluid of density ρ. The cylinder piston is given an acceleration a. The resulting force, due the inertia of the fluid in the line, F is

$$F = A_0/A_1 \times L \times \rho \times \alpha$$

It can be seen that the inertia force is sensitive to fluid density (generally fixed for hydraulic fluids), line length and, line area and very sensitive to piston area. Any reduction in line length and increase in line area will reduce the fluid inertia effects. It is in practice more convenient to reduce the line length rather than increasing the line area by increasing the diameter of the fluid conduits. The latter change can lead to packaging difficulties under the vehicle because of the limited space available for installing the fluid conduits. Another beneficial change which can reduce fluid inertia effects is to increase the mechanical advantage (or lever ratio) from the wheels to the fluid cylinders. This can lead to higher peak pressures, but lower fluid accelerations.

The modes likely to be influenced by high frequency inputs are single wheel input and two wheel parallel bump input. In the roll control system layout shown in FIGS. **1** to *6j*, for a two wheel parallel bump input, fluid is required to travel from the front left upper chamber **51** to the front right lower chamber **52** along with fluid travel from the front right upper chamber **51** to the front left lower chamber **52**. There is a minor flow into the accumulator **16** (see FIGS. *6d* and *6e*).

For a single wheel input fluid must travel from the cylinder chambers directly to the accumulator (see FIGS. *6a* to *6c*).

With the fluid control valve **26** open, the two wheel parallel bump input flows are unchanged. For a single wheel input the flow now passes through the fluid control valve **26** with little flow to the accumulators **16**.

In the above situations in the above-described roll control system layout, the fluid must travel down lines of a reasonable length and reasonable diameter. This provides a significant inertia effect.

As the system provides roll support and must provide a suitable roll moment distribution the front cylinders are generally a larger diameter than the rear. Due to the sensitivity to piston area the front is more likely to show fluid inertia effects than the rear.

The piston areas are further fixed by maximum required operating pressures.

Another possible arrangement of the roll control system is shown in FIG. **7**. The layout of the fluid circuits **7** is varied to provide for the shortest route from one front cylinder **8** to the other reducing line length and hence fluid inertia effects.

In particular, the upper chamber **61** of each adjacent pair of front wheel cylinders **8** are connected by a respective fluid conduit **70** to the lower chamber **52** of the adjacent front wheel cylinder **8**. These fluid conduits **70** are connected to the corresponding fluid conduits **70** of the rear wheel cylinders by longitudinal fluid conduits **71** to provide a pair of fluid conduits of the system. During single wheel and two wheel bounce, much of the fluid is transferred between the fluid chambers **51** and **52** of each pair of front wheel cylinders **8** and/or between the fluid chambers **51** and **52** of each pair of rear wheel cylinders **8**.

Only a relatively small amount of fluid need pass through the longitudinal fluid conduits **71** where inertia effects are likely to be more pronounced. The opening of the flow control valve **26**, as noted previously, results in short circuiting of the fluid circuits such that fluid is caused to flow between the upper and lower chambers **51** and **52** of the wheel cylinders **8**.

Because the fluid inertia effects are likely to be more pronounced at the front as noted previously, accumulators may be provided on each of the fluid conduits **70** connecting the front wheel cylinder chambers **51** and **52**. The accumulators **16** act to accommodate a large fluid flow resulting from a single wheel input when the flow control valve **26** is closed.

FIG. **8** shows a variation of the arrangement shown in FIG. **6**, with further accumulators **16** being provided on each of the fluid conduits **70** connecting the rear fluid cylinder chambers **51** and **52**.

Added accumulators **16** at the rear fluid conduits **70** allow fluid to generally bypass the longitudinal fluid conduit **70** resulting in less effective line length and reduced inertia effects.

The preferred embodiment of the roll control system shown in FIG. **9** is similar to the arrangement shown in FIGS. **1** and **2** except that the single bridge passage **20** and flow control valve **26** is replaced with a respective bridge passage **20** and flow control valve **26** for each wheel cylinder **8**. This allows the fluid flow for each wheel cylinder **8** to be independently short circuited to allow for relatively direct flow between the upper and lower chambers **51** and **52** of the wheel cylinders **8**.

Providing four separate bridge passages **20** and flow control valves located at each of the cylinders **8** therefore allows for a direct short circuit of the system. Most of the flow is bypassed directly around each cylinder **8** through a short and reasonable area line for all inputs with the flow control valve **26** open. This provides a significant reduction in the fluid inertia effects. The damping is however still maintained as the dampers **15**,**18** are in this fluid loop. Operation with the four flow control valves **28** closed will not however offer any fluid inertia improvements.

The flow control valves 26 could be digital (on or off only), multi-position or proportional depending on the level of damping control required. The valve 26 must seal when fully closed.

FIGS. 10*a* to 10*c* together illustrate another preferred embodiment of the roll control system similar to the embodiment shown in FIG. 1, but where the bridge passage 20 and roll control valve 26 is omitted (see FIG. 10*c*). Each wheel cylinder 8 is however adapted to include a fluid flow control assembly which allows for direct fluid flow between the upper and lower chambers 51 and 52 of each wheel cylinder 8.

The pistons 80 of each wheel cylinder have a control valve 81 inserted therein (see FIGS. 10*b* and 10*c*). The control valve 81 controls the rate of fluid through a piston passage 82 providing for fluid communication between the upper and lower chambers 61, 52 of the wheel cylinder 8. A rotary valve 81 is shown but any design is applicable. This rotary valve 81 is rotatable by a shaft 83 passing through the piston rod 84 between an open position (FIG: 10*b*) and a closed position (FIG. 10*c*). Fluid flow between the upper and lower chambers 51, 52 is allowed when the valve 81 is open. This valve 81 directly connecting the upper and lower chamber of each cylinder thereby provides a short fluid path. Again fluid inertia effects are significantly reduced. An in-line damper 85 is required to damp the fluid tow through the piston passage 82 to thereby damp the wheel movement, as the dampers in the fluid conduits have effectively been bypassed. The valves 81 could be digital (on or off only), multi-position or proportional depending on the level of damping control required in comfort mode. The valve 81 must seal when fully closed. The construction of the valve may be different to that illustrated, such as a disc with holes arranged in it and attached to the shaft 83. The piston may have spring steel shims on the holes on either side of the piston to provide damping control. The holes in the disc may be in the form of tapered slots to provide variable flow areas and therefore a degree of variable damping control.

FIG. 11 shows a preferred embodiment of the roll control system which utilises the fluid conduit layout of the system shown in FIG. 7, but further includes a respective flow control valve 26 for each wheel cylinder.

This layout minimises fluid inertia effects even when the flow control valves 26 are closed and provides a further reduction of fluid inertia effects with the flow control valves 28 open. It should however be appreciated that the flow control valves 26 could alternatively be inside the pistons 80 as shown in FIGS. 10*b* and 10*c*.

The preferred embodiment shown in FIG. 12 is the same as the arrangement shown in FIG. 11 but with additional rear accumulators 16 for the same reasons as in the arrangement shown in FIG. 8.

What is claimed is:

1. A damping and roll control system for a vehicle suspension system, the vehicle having at least one pair of laterally spaced front wheel assemblies and at least one pair of laterally spaced rear wheel assemblies, each wheel assembly including a wheel and a wheel mounting locating the wheel to permit movement of the wheel in a generally vertical direction relative to a body of the vehicle, and vehicle support means for providing at least substantially a major portion of the support for the vehicle; the damping and roll control system including:

wheel cylinders respectively locatable between each wheel mounting and the body of the vehicle, each wheel cylinder including an inner volume separated into first and second chambers by a piston supported within the wheel cylinder;

first and second fluid circuits respectively providing fluid communication between the wheel cylinders by fluid conduits, each of said fluid circuits providing fluid communication between the first chambers of the wheel cylinders on one side of the vehicle and the second chambers of the wheel cylinders on the opposite side of the vehicle to thereby provide roll support decoupled from a warp mode of the vehicle suspension system by providing a roll stiffness about a level roll attitude whilst simultaneously providing substantially zero warp stiffness;

each fluid circuit including one or more fluid accumulators for providing roll resilience;

the or at least one of the accumulators on each fluid circuit including an accumulator damping means for controlling the rate of fluid into and out of the accumulator;

damper means for controlling the rate of fluid flow out of or into at least one chamber of each wheel cylinder; and

selection means for selectively providing fluid communication between the first and second fluid circuits;

the damping and roll control system thereby providing substantially all of the damping of the vehicle suspension system.

2. A damping and roll control system according to claim 1, wherein the vehicle support means provides at least substantially all of the support for the vehicle.

3. A damping and roll control system according to claim 1, wherein the vehicle support means for at least one end of the vehicle include first support means, the first support means supporting at least a portion of the load on the wheels and providing substantially zero roll and warp stiffness.

4. A damping and roll control system according to claim 1, wherein the fluid circuit includes a first fluid conduit providing fluid communication between the first chambers of the wheel cylinders on one side of the vehicle, and a second fluid conduit providing fluid communication between the second chambers of the wheel cylinders on the opposite side of the vehicle, the first and second fluid conduits being in fluid communication.

5. A damping and roll control system according to claim 1, wherein each fluid circuit includes first and second diagonal fluid conduits, each respectively providing fluid communication between the first chamber of one wheel cylinder on one side of the vehicle and the second chamber of the diagonally opposite wheel cylinder on the other sides of the vehicle, the first diagonal fluid conduit between one pair of diagonally opposite wheel cylinders being in fluid communication with the second diagonal fluid conduit between the other pair of diagonally opposite wheel cylinders.

6. A damping and roll control system according to claim 1, wherein each fluid circuit includes a front fluid conduit providing fluid communication between the wheel cylinders of the front wheel assemblies, and a rear fluid conduit providing fluid communication between the wheel cylinders of the rear wheel assemblies, with the front and read conduits respectively providing fluid communication between the first chamber of the wheel cylinder at one side of the vehicle with the second chamber of the wheel cylinder at the opposite side of the vehicle, the front and rear conduits being in fluid communication.

7. A damping and roll control system according to claim 1, wherein the damper means are located at or in the wheel cylinders.

8. A damping and roll control system according to claim 1, wherein the damper means are located in the conduits.

9. A damping and roll control system according to claim 1, wherein the damper means are located in a manifold block providing fluid communication between the first and second conduits to form the first and second fluid circuits.

10. A damping and roll control system according to claim 1, wherein the dampers means are bi-directional damper valves for controlling the fluid flow rate to and from at least one of the first or second chambers of each said wheel cylinder.

11. A damping control system according to claim 1, further including bypass means for bypassing the damper means.

12. A damping and roll control system according to claim 1, wherein the damper means includes a single direction damper valve for controlling the fluid flow rate from each said chamber of each said wheel cylinder.

13. A damping and roll control system according to claim 12, wherein the single direction damper valve is used in parallel with a non-return valve.

14. A damping and roll control system according to claim 1, wherein each fluid circuit includes a second fluid accumulator, and a roll resilience switching valve located between the second accumulator and the fluid circuit for selectively communicating the second accumulator with said fluid circuit and thereby control the degree of roll resilience.

15. A roll control system according to claim 1, wherein a bypass passage is provided around the damper means for the accumulator, the bypass passage including a valve to open or close the bypass passage.

16. A damping and roll control system according to claim 1, wherein the roll control system has a pressure precharge.

17. A damping and roll control system according to claim 1, wherein each wheel cylinder associated with at least one of said pair of laterally spaced wheel assemblies includes a piston rod extending respectively from opposing sides of the piston, the diameter of each piston rod being at least substantially equal, such that the effective piston area in the first and second chamber of each wheel cylinders are at least substantially equal.

18. A damping and roll control system according to claim 1, wherein each wheel cylinder associated with at least one of said parts of laterally spaced wheels includes a hollow piston rod and an inner rod said hollow piston rod extending from one side of the piston through one chamber of the cylinder, said inner rod being supported within the other chamber of the wheel cylinder and at least partially accommodate within the hollow piston rod to form a rod chamber, the cylinder having a first annular chamber, a second annular chamber and the rod chamber.

19. A damping and roll control system according to claim 18, wherein the first chamber of the damping and roll control system is the first annular chamber and the second chamber of the damping and roll control system is the second annular chamber.

20. A damping and roll control system according to claim 18, wherein the first chamber of the damping and roll control system is the rod chamber and the second chamber of the damping and roll control system is the second annular chamber.

21. A damping and roll control system according to claim 20, wherein the second annular chamber has a larger effective piston area than the rod chamber.

22. A damping and roll control system according to claim 20, wherein the second annular chamber has the same effective piston area as the rod chamber.

23. A damping and roll control system according to claim 20, wherein the first annular chamber of the wheel cylinder forms part of a support circuit for the vehicle.

24. A damping and roll control system according to claim 1, wherein said selection means includes at least one bridge passage connecting said first and second fluid circuits, and a flow control valve for controlling the flow through the bridge passage.

25. A damping and roll control system according to claim 24, further including an accumulator on the bridge passage.

26. A damping and roll control system according to claim 24, wherein a respective bridge passage and flow control valve is provided for each wheel cylinder.

27. A damping and roll control system according to claim 1, wherein the piston of each wheel cylinder includes an integral flow control valve and damper valve for controlling the flow between the first and second chambers.

28. A method of controlling the roll damping and roll stiffness of a damping and roll control system for a vehicle suspension system, the damping and roll control system including:

wheel cylinders respectively locatable at wheel assemblies of the vehicle, each wheel cylinder including an inner volume separated into first and second chambers by a piston supported within the wheel cylinder;

first and second fluid circuits respectively providing fluid communication between the wheel cylinders by fluid conduits, each of said fluid circuits providing fluid communication between the first chambers of the wheel cylinders on one side of the vehicle and the second chambers of the wheel cylinders on the opposite side of the vehicle to thereby provide roll support decoupled from a warp mode of the vehicle suspension system by providing a roll stiffness about a level roll attitude whilst simultaneously providing substantially zero warp stiffness;

each fluid circuit including one or more fluid accumulators for providing roll resilience;

the or at least one of the accumulators on each fluid circuit including an accumulator damper means for controlling the rate of fluid flow into and out of the accumulator;

damper means for controlling the rate of fluid flow into and out of at least one chamber of each wheel cylinder; and

selection means for selectively providing fluid communication between the first and second fluid circuits;

the method including opening the selection means to provide fluid communication between the first and second fluid circuits when the damping and roll system is required to provide a relatively low level of roll stiffness and roll damping; and

closing the selection means to prevent fluid communication between the first and second fluid circuits when the damping and roll system is required to provide a relatively high level of roll stiffness and roll damping.

29. A method according to claim 28, including opening the selection means to bypass the fluid flow through at least a substantial portion of the fluid circuits when there is a single wheel input or two wheel parallel bump input to the damping and roll control system.

30. A method according to claim 28, including opening the selection means to entirely bypass the fluid flow from the fluid conduits at said predetermined wheel inputs.

* * * * *

UNITED STATES PATENT AND TRADEMARK OFFICE
CERTIFICATE OF CORRECTION

PATENT NO. : 6,761,371 B1
DATED : July 13, 2004
INVENTOR(S) : Christopher B. Heyring et al.

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

Column 2,
Line 23, "6,684,496" should be -- 5,584,498 --.

Column 3,
Line 66, "charmer" should be -- chamber --.

Column 4,
Line 22, "alterative" should be -- alternatively --.

Column 5,
Line 25, "uniaden" should be -- unladen --.
Line 31, "not" should be -- net --.

Column 8,
Line 35, "and (or" should be -- and/or --.

Column 9,
Line 59, "lamping" should be -- damping --.

Column 11,
Line 25, "Applicants" should be -- Applicants' --.
Line 30, "cylinder B" should be -- cylinder 8 --.

Column 12,
Lines 26 and 37, delete "which".
Line 38, "dummy rod 60" should be -- dummy rod 61 --.
Lines 45 and 61, "roil" should be -- roll --.
Line 47, "chamber 83" should be -- chamber 63 --.
Line 48, "dummy rod 82" should be -- dummy rod 61 --.

Column 13,
Line 30, "of" should be -- or --.

Column 14,
Line 65, "circulating" should be -- circuiting --.

UNITED STATES PATENT AND TRADEMARK OFFICE

CERTIFICATE OF CORRECTION

PATENT NO. : 6,761,371 B1
DATED : July 13, 2004
INVENTOR(S) : Christopher B. Heyring et al.

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

<u>Column 15,</u>
Line 12, "state" should be -- static --.
Line 13, "forcas" should be -- forces --.
Line 23, "lenghts" should be -- lengths --.
Line 37, after "due" insert -- to --.
Line 38, "A_o" should be -- A_o^2 --.

<u>Column 16,</u>
Line 16, "upper chamber 61" should be -- upper chamber 51 --.

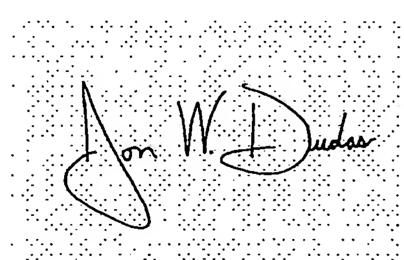
<u>Column 17,</u>
Line 25, "tow" should be -- flow --.
Lines 44-45, "valves 28" should be -- valves 26 --.

<u>Column 18,</u>
Line 57, "read" should be -- rear --.

<u>Column 19,</u>
Line 6, "dampers" should be -- damper --.
Lines 47-48, "accommodate" should be -- accommodated --.

Signed and Sealed this

Thirtieth Day of August, 2005



JON W. DUDAS
Director of the United States Patent and Trademark Office

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(12) **United States Patent**
Kobayashi

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(54) **SUSPENSION SYSTEM FOR MOTOR VEHICLE**

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B60G 21/00 (2006.01)

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(58) **Field of Classification Search** 280/124.104, 280/124.106, 124.158, 124.159, 124.161, 280/5.505, 5.506, 5.507, 124.15; 267/186, 267/64.28

See application file for complete search history.

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(57) **ABSTRACT**

A vehicle suspension system of a motor vehicle includes a plurality of suspension devices mounted on the vehicle with respect to right and left wheels of the vehicle, respectively, a first behavior controller that controls the motion of each of the suspension devices when a vehicle body undergoes a first behavior, and a second behavior controller that controls the motion of each of the suspension devices when the vehicle body undergoes a second behavior, independently of the first behavior controller.

20 Claims, 7 Drawing Sheets

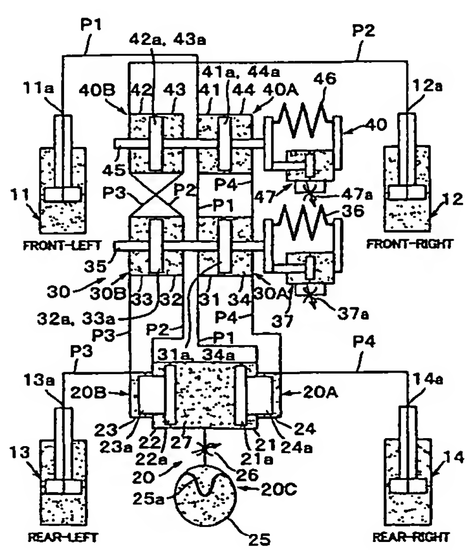
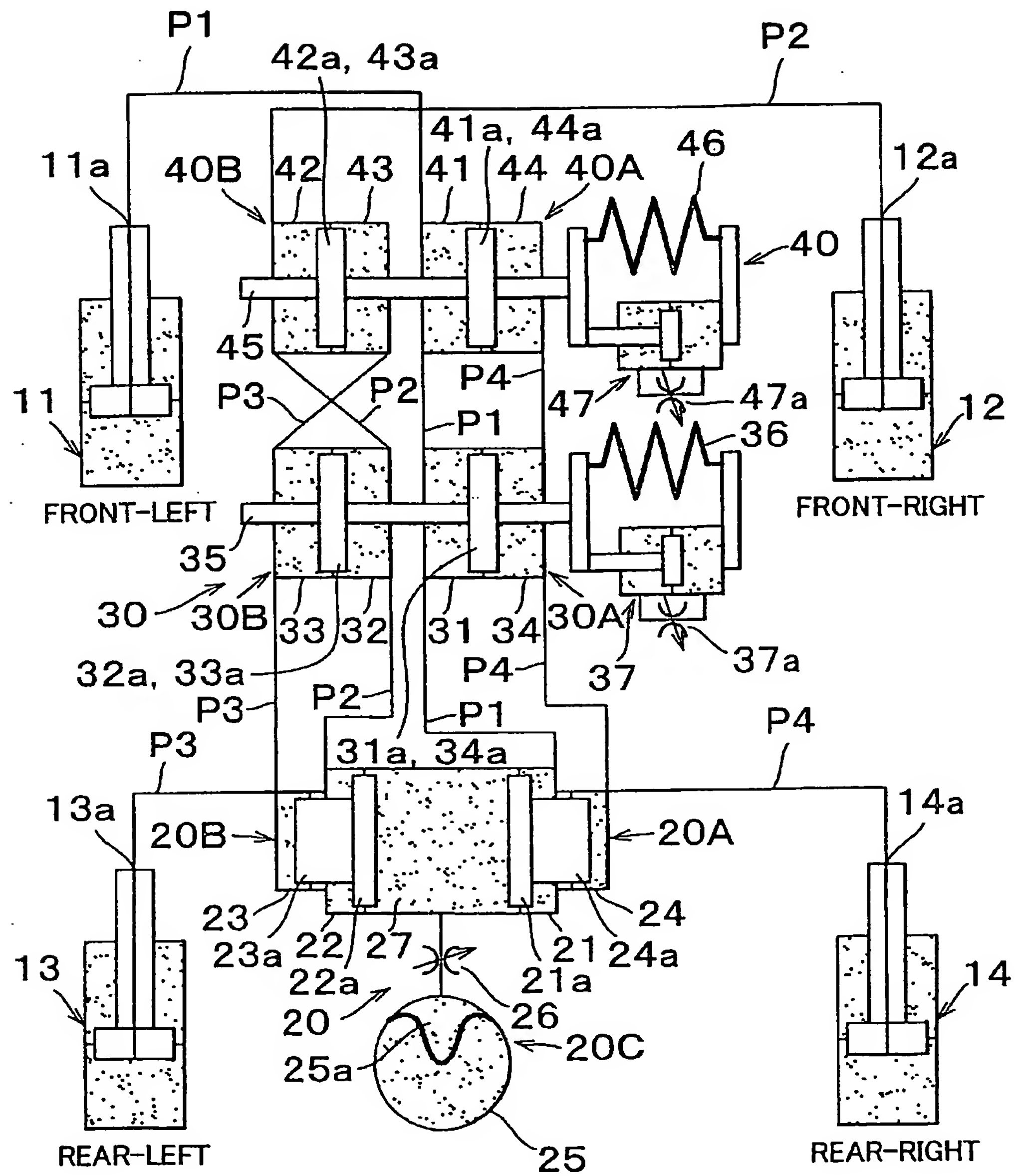


FIG. 1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P3
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P1
47
47a
12
36
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35
30
30B
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32
31
34
30A
37
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P4
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23a
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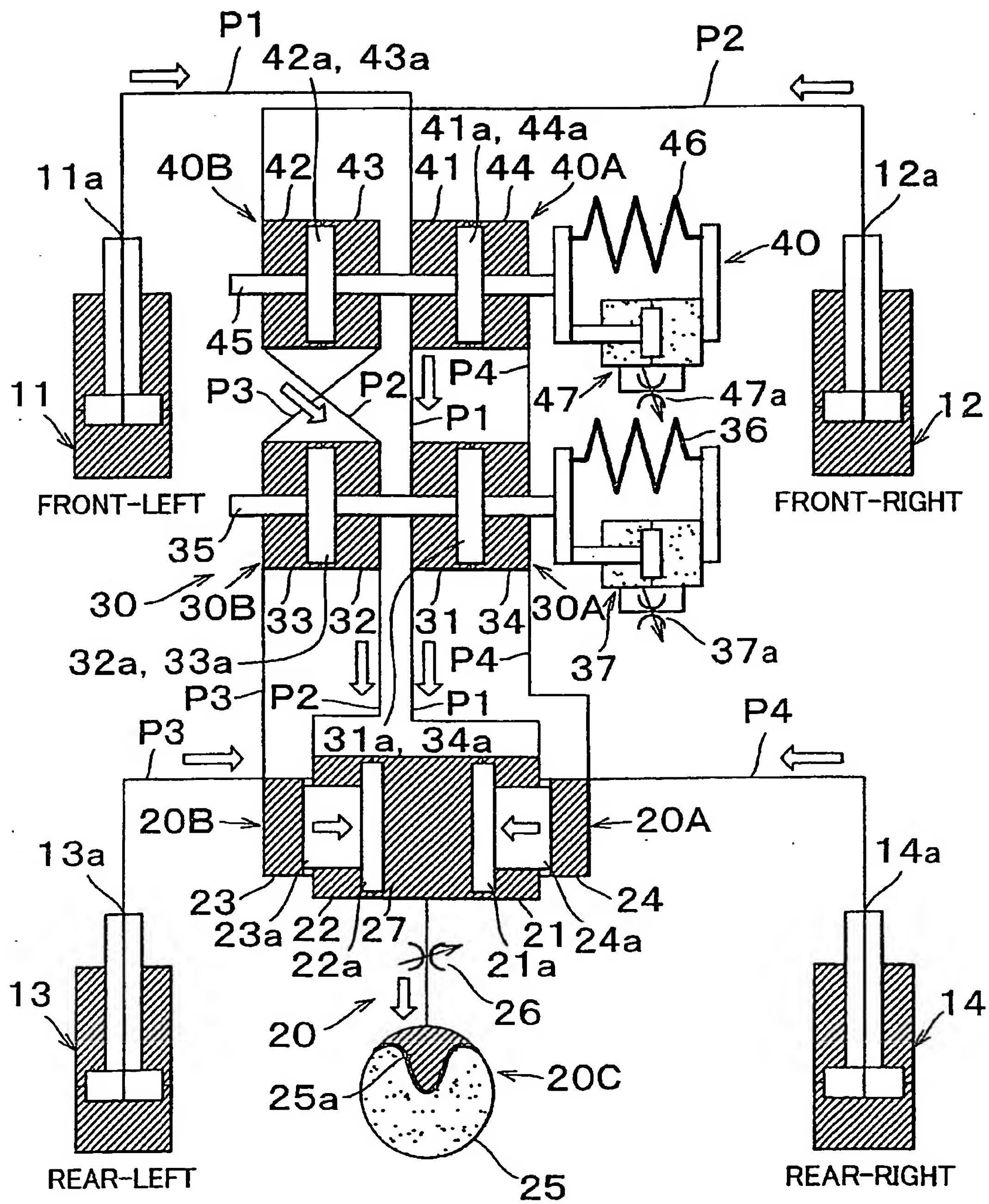
REAR-RIGHT

FIG. 2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12
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30B
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P3
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P1
P3
P4
31a, 34a
20B
20A
13a
14a
23
24
23a
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21a
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14
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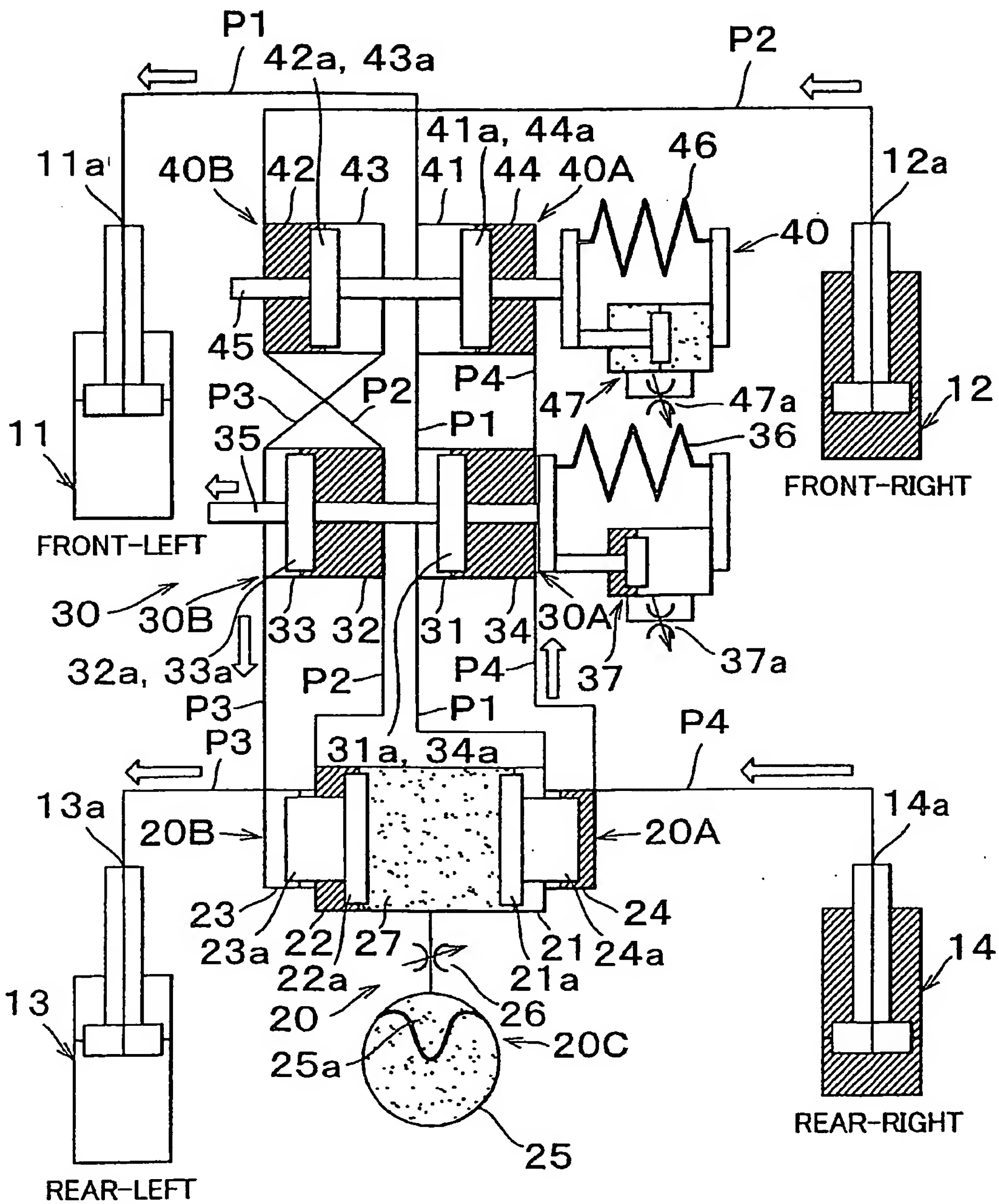
REAR-LEFT
REAR-RIGHT
25

FIG. 3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P2
P4
47
47a
12
11
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P1
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FRONT-RIGHT
FRONT-LEFT
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30B
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31
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30A
37a
32a, 33a
P2
P4
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P3
P1
P4
P3
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13a
14a
20B
20A
23
24
23a
22
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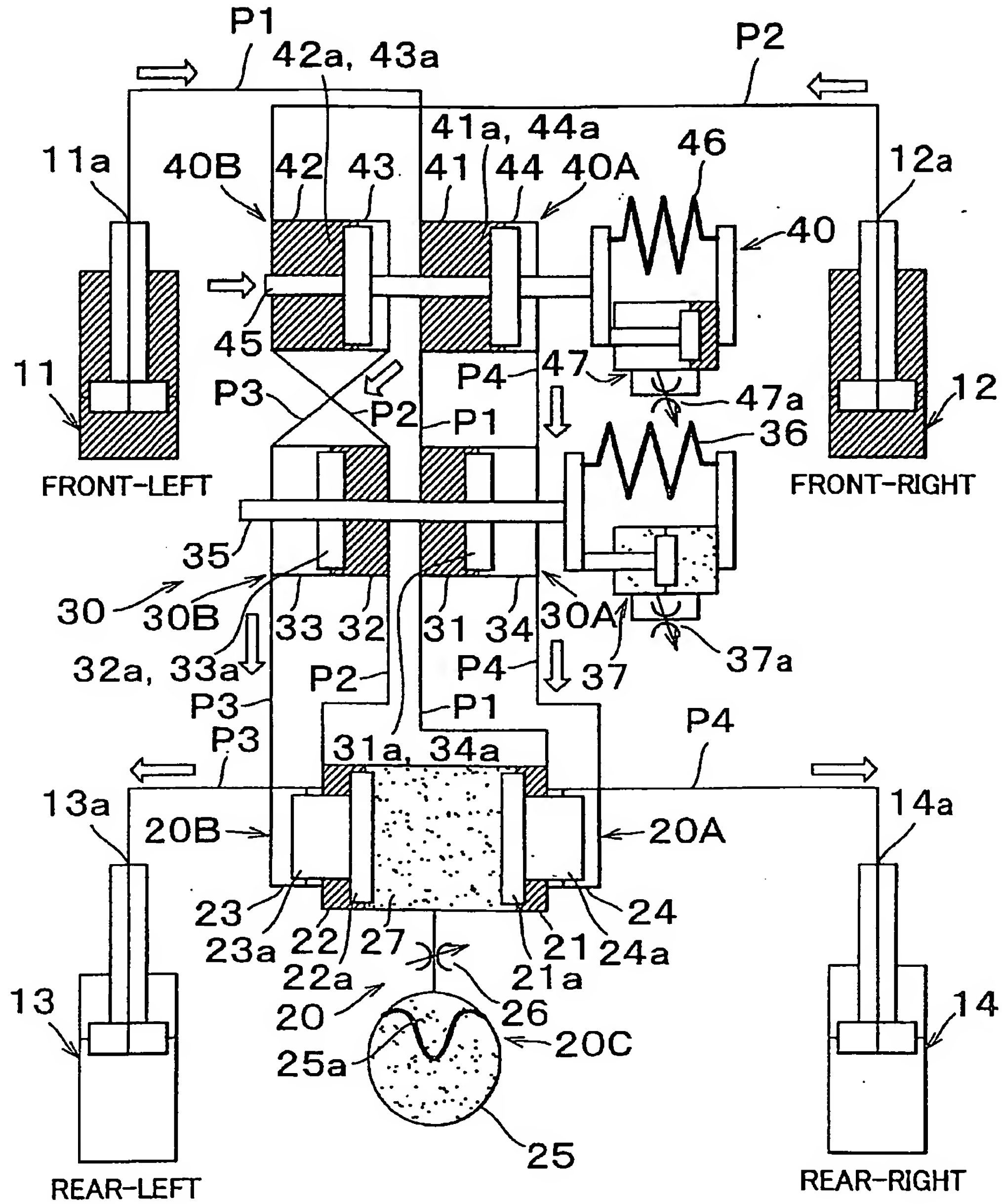
14
22a
21a
13
20
26
20C
25a
25
REAR-RIGHT
REAR-LEFT

FIG. 4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P4
P3
P2
P1
47a
12
36
FRONT-LEFT
FRONT-RIGHT
35
30
30B
33
32
31
34
30A
37a
32a, 33a
P2
P4
37
P3
P1
P3
31a, 34a
P4
13a
14a
20B
20A
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24
23a
22
27
21
24a
22a
21a
13
20
26
14
25a
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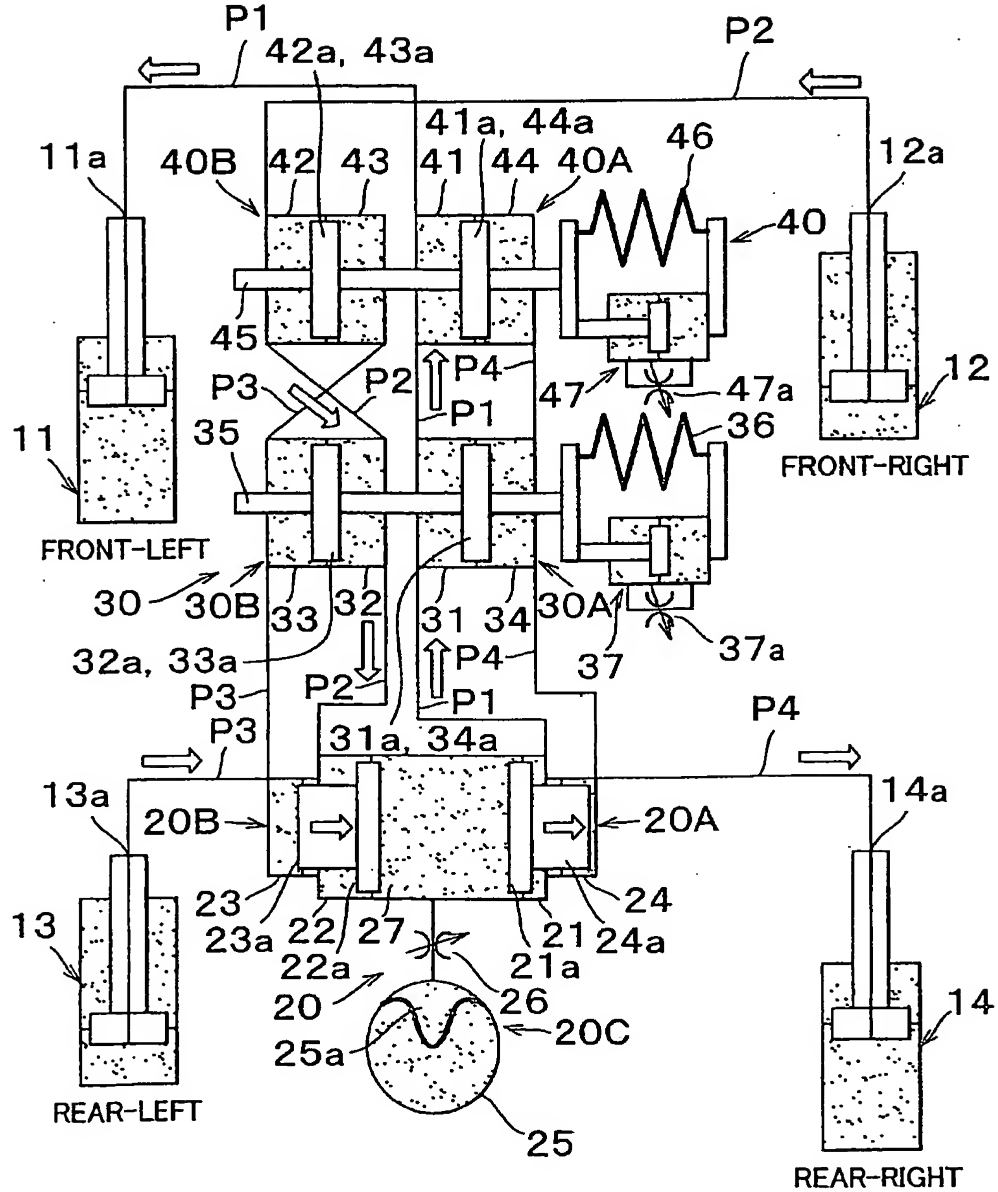
REAR-LEFT
REAR-RIGHT

FIG. 5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P2
P1
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47a
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36
FRONT-RIGHT
FRONT-LEFT
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30B
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32
31
34
30A
37a
32a, 33a
P4
37
P3
P2
P1
P3
31a, 34a
P4
13a
20B
20A
14a
23
24
13
23a
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21
24a
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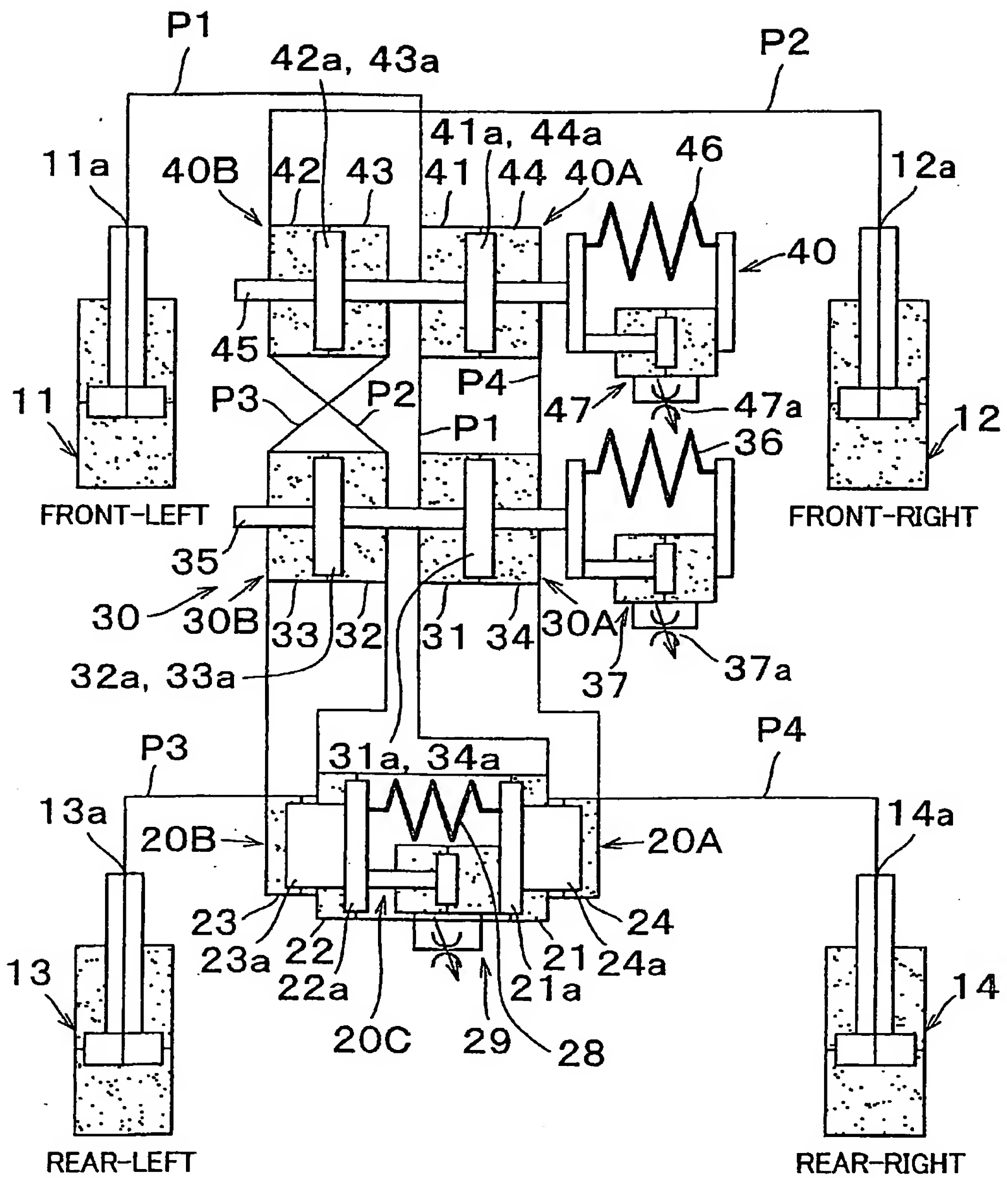
21a
20
26
20C
25a
14
REAR-LEFT
25
REAR-RIGHT

FIG. 6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P3
P2
P4
P1
47
47a
36
12
FRONT-LEFT
FRONT-RIGHT
35
30
30B
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32
31
34
30A
37a
32a, 33a
37
P3
31a, 34a
P4
13a
20B
20A
14a
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23a
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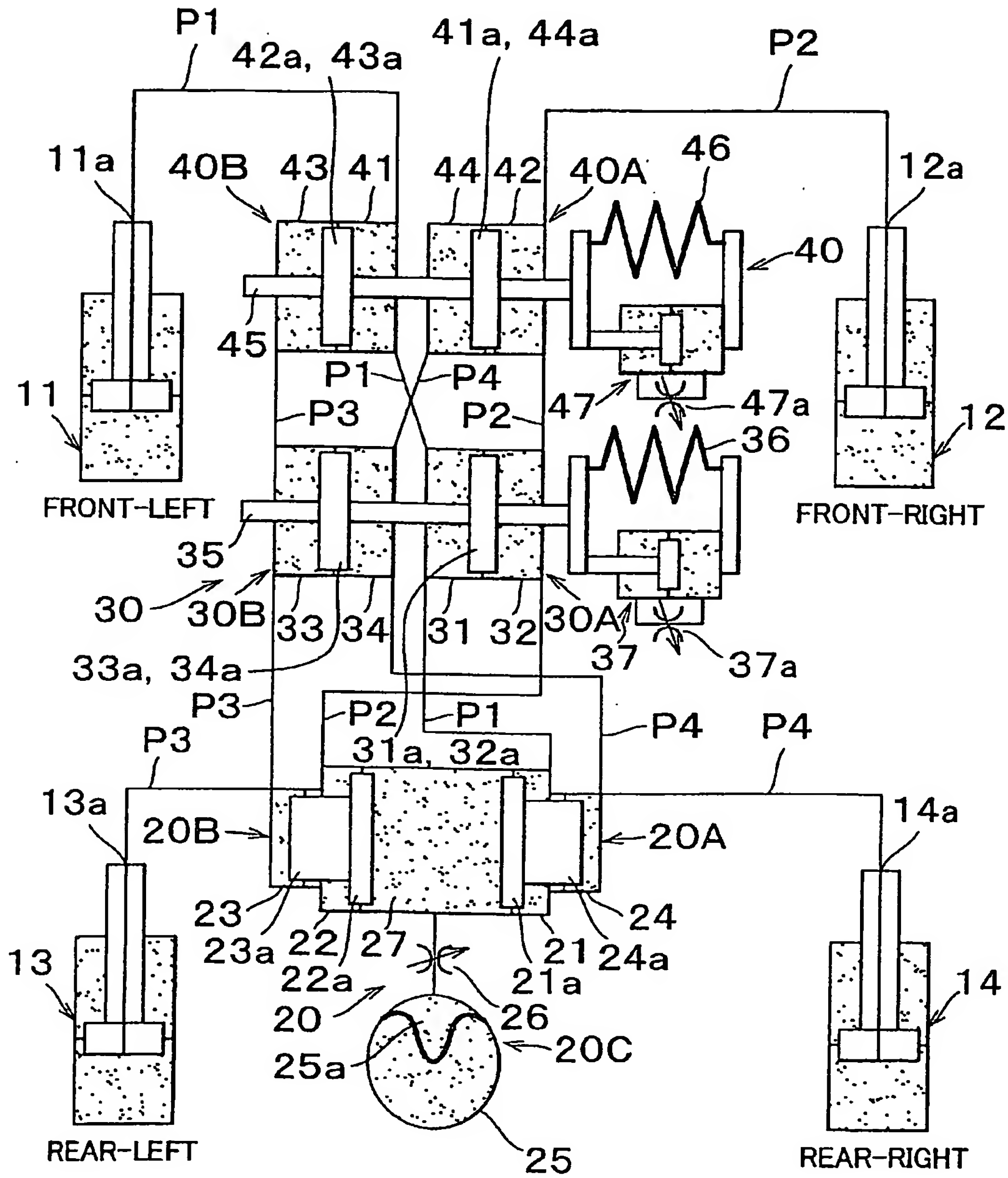
22a
21a
14
20C
29
28
REAR-LEFT
REAR-RIGHT

FIG. 7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44
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40A
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12a
40
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P1
P4
P3
P2
47
47a
11
36
12
FRONT-LEFT
FRONT-RIGHT
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30B
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31
32
30A
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37a
33a, 34a
P3
P2
P1
P4
P4
P3
31a, 32a
13a
20B
20A
14a
23
24
23a
22
27
21
24a
22a
21a
13
14
20
26
20C
25a
25
REAR-LEFT
REAR-RIGHT

SUSPENSION SYSTEM FOR MOTOR VEHICLE

BACKGROUND OF THE INVENTION

1. Field of the Invention

The invention relates to a vehicle suspension system employed in a motor vehicle, such as a four-wheel vehicle, and in particular to a vehicle suspension system that includes vehicle behavior controllers that control the motion of each suspension device (e.g., hydraulic cylinder for suspension) mounted with respect to each of all right and left wheels, and is adapted to control or suppress particular types of behavior, such as bouncing, pitching and rolling, of a vehicle body.

2. Description of Related Art

An example of a vehicle suspension system of the above type is disclosed in, for example, Japanese laid-open Patent Publication No. 6-509997. In the vehicle suspension system disclosed in this publication, suspension hydraulic cylinders respectively mounted with respect to front and rear, right and left wheels of the vehicle are diagonally connected to each other by hydraulic pipes, so that the suspension system can suppress pitching and rolling of the vehicle body while assuring sufficient road-holding characteristic of the wheels on an unleveled ground.

The operating characteristics of the vehicle suspension system as shown in the above-indicated publication are determined by accumulators (e.g., gas springs) connected to and communicating with respective hydraulic pipes for connecting diagonally located cylinders, or accumulators connected to and communicating with the hydraulic pipes for connecting diagonally located cylinders and a single load distribution unit, or an actuator that includes a single load distribution unit to which the hydraulic pipes for connecting diagonally located cylinders are connected for fluid communication. With this arrangement, the operating characteristics of the suspension system cannot be individually or separately set for suppressing each type of vehicle behaviors, such as pitching and rolling, of the vehicle body. Thus, it has been difficult to set the operating characteristics of the suspension system suitable for each type of vehicle behavior.

In addition, the known vehicle suspension system as described above is constructed such that a pair of hydraulic chambers formed on the opposite sides of a piston in each of the suspension hydraulic cylinders are diagonally connected to a pair of hydraulic chambers formed in a corresponding suspension hydraulic cylinder that is located in a diagonal relationship with the above-indicated hydraulic cylinder. Namely, two ports formed in each of the suspension hydraulic cylinders are diagonally connected to two ports formed in the corresponding suspension hydraulic cylinder, to thus provide two hydraulic pipe systems. Thus, the hydraulic pipes are arranged in a complicated manner, which results in increased cost and weight of the pipes, and eventually those of the suspension system. While the vehicle suspension system as described above is able to suppress pitching and rolling of the vehicle body, the system is not able to suppress bouncing (i.e., behavior in the heaving direction) of the vehicle body.

SUMMARY OF THE INVENTION

In view of the above situations, the invention provides a vehicle suspension system of a motor vehicle, which includes a plurality of suspension devices mounted on the vehicle with respect to right and left wheels of the vehicle, respectively, a first behavior controller that controls a motion of each of the suspension devices when a vehicle body undergoes a first behavior, and a second behavior controller that controls a motion of each of the suspension devices when the vehicle body undergoes a second behavior, independently of the first behavior controller. In this case, the first behavior and the second behavior may be selected from bouncing, rolling and pitching of the vehicle body.

In the vehicle suspension system according to the above aspect of the invention, the motion of each of the suspension devices mounted on the vehicle with respect to all of the right and left wheels is controlled independently by the first behavior controller and the second behavior controller, and therefore the characteristics of the behavior controllers can be separately or individually set for respective types of behaviors. Thus, the characteristics of each behavior controller are independently set to those suitable for controlling or suppressing each type of behavior (e.g., bouncing, rolling or pitching), whereby each type of behavior can be optimally controlled or suppressed.

According to another aspect of the invention, the invention provides a vehicle suspension system of a motor vehicle, which includes (a) a plurality of suspension hydraulic cylinders mounted on the vehicle with respect to front-right, front-left, rear-right, rear-left wheels of the vehicle, each of the suspension hydraulic cylinders having a single port, and (b) a plurality of control hydraulic cylinders each of which is connected to the single port of a corresponding one of the suspension hydraulic cylinders via a pipe, for controlling a motion of the corresponding suspension hydraulic cylinder. In the vehicle suspension system, a pair of diagonal hydraulic control cylinders are provided by connecting the control hydraulic cylinders such that hydraulic pressures in two of the control hydraulic cylinders connected to diagonally located ones of the suspension hydraulic cylinders change in substantially the same way, and the pair of diagonal hydraulic control cylinders are opposed to each other and coupled by a coupling device capable of controlling motions of the diagonal hydraulic control cylinders.

In the vehicle suspension system according to the above aspect of the invention, a hydraulic circuit is constructed by connecting the single ports of the respective suspension hydraulic cylinders mounted for the front and rear, right and left wheels, to the corresponding control hydraulic cylinders, via respective pipes. Thus, the hydraulic circuit can be simply constructed at a relatively low cost. Furthermore, the vehicle suspension system is able to not only effectively suppress the behavior (bouncing) of the vehicle body in the heaving direction, but also suitably deal with the situation where a force that twists the vehicle body is applied to the front and rear, right and left wheels when the vehicle is running on an unleveled ground, for example. More specifically, when the vehicle body twists on an unleveled ground, the pair of diagonal hydraulic control cylinders are freely operated in the same phase or direction, and therefore the vertical load and driving force measured at each wheel are less likely to be reduced. Thus, the vehicle suspension system permits the vehicle posture or attitude to be favorably maintained while assuring sufficient driving force of each wheel, without making the hydraulic circuit undesirably complicated.

BRIEF DESCRIPTION OF THE DRAWINGS

The foregoing and/or further objects, features and advantages of the invention will become more apparent from the

following description of exemplary embodiments with reference to the accompanying drawings, in which like numerals are used to represent like elements and wherein:

FIG. 1 is a hydraulic circuit diagram schematically showing one exemplary embodiment of a vehicle suspension system of the invention;

FIG. 2 is a view showing an operating state of the vehicle suspension system as shown in FIG. 1 at the time of bouncing of the vehicle body;

FIG. 3 is a view showing an operating state of the vehicle suspension system as shown in FIG. 1 at the time of rolling of the vehicle body (e.g., when the vehicle turns left);

FIG. 4 is a view showing an operating state of the vehicle suspension system as shown in FIG. 1 at the time of pitching of the vehicle body (e.g., when the vehicle dives);

FIG. 5 is a view showing an operating state of the vehicle suspension system as shown in FIG. 1 at the time of twisting of the vehicle;

FIG. 6 is a hydraulic circuit diagram schematically showing another embodiment of a vehicle suspension system of the invention; and

FIG. 7 is a hydraulic circuit diagram schematically showing a further embodiment of a vehicle suspension system of the invention.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENT(S)

One exemplary embodiment of the invention will be described in detail with reference to the accompanying drawings. FIG. 1 schematically shows a suspension system of a motor vehicle according to the embodiment of the invention. In the suspension system, each of suspension hydraulic cylinders **11**, **12**, **13**, **14** is connected to a bouncing controller **20**, a rolling controller **30** and a pitching controller **40**, via respective pipes P1, P2, P3, P4. The suspension hydraulic cylinders **11**, **12**, **13**, **14** are mounted on the vehicle with respect to front and rear, right and left wheels, and have single ports **11***a*, **12***a*, **13***a*, **14***a*, respectively.

The bouncing controller **20** functions to control the motions of the suspension hydraulic cylinders **11**, **12**, **13**, **14** when bouncing as one type of vehicle behavior occurs, and includes bouncing control cylinders **21**, **22**, **23**, **24** that are respectively connected to the ports **11***a*, **12***a*, **13***a*, **14***a* of the suspension hydraulic cylinders **11**, **12**, **13**, **14** via the pipes P1, P2, P3, P4, respectively. The bouncing control cylinders **21**, **22**, **23**, **24** include respective pistons **21***a*, **22***a*, **23***a*, **24***a*, which have substantially the same pressure-receiving area.

The bouncing control cylinders **21**, **24** are respectively connected to the suspension hydraulic cylinders **11**, **14** that are located diagonally in the vicinity of the front, left wheel and the rear, right wheel, such that the hydraulic pressures in the cylinders **21**, **24** change in the same way. Namely, the pistons **21***a*, **24***a* in the bouncing control cylinders **21**, **24** move in the same direction in accordance with increases and decreases in the hydraulic pressures applied thereto. The bouncing control cylinders **21**, **24** constitute a diagonal hydraulic control cylinder **20**A, in which the pistons **21***a*, **24***a* of the bouncing control cylinders **21**, **24** are coupled to each other and are formed as an integral assembly or unit.

On the other hand, the bouncing control cylinders **22**, **23** are respectively connected to the suspension hydraulic cylinders **12**, **13** located diagonally in the vicinity of the front, right wheel and the rear, left wheel, such that the hydraulic pressures in the cylinders **22**, **23** change in the same way. Namely, the pistons **22***a*, **23***a* in the bouncing control cylinders **22**, **23** move in the same direction in accordance with increases and decreases in the hydraulic pressures applied thereto. The bouncing control cylinders **22**, **23** constitute a diagonal hydraulic control cylinder **20**B, in which the pistons **22***a*, **23***a* of the bouncing control cylinders **22**, **23** are coupled to each other and are formed as an integral assembly or unit.

The diagonal hydraulic control cylinders **20**A, **20**B are constructed in symmetric shape, and are disposed opposite to each other. The cylinders **20**A, **20**B are connected to a coupling device **20**C that is operable to control the motions of the diagonal hydraulic cylinders **20**A, **20**B. The coupling device **20**C includes an accumulator **25** that functions as a spring element and is operable by using, for example, a gas or a spring. The coupling device **20**C is a liquid-tight coupling structure using a hydraulic fluid as a medium, and is provided with a hydraulic chamber **27** that communicates with a hydraulic chamber **25***a* of the accumulator **25** via a variable restrictor **26** that functions as a damping element for suppressing vibration of the spring element.

The rolling controller **30** functions to control the motions of the suspension hydraulic cylinders **11**, **12**, **13**, **14** when rolling as another type of vehicle behavior occurs, and includes rolling control cylinders **31**, **32**, **33**, **34** that are respectively connected to the ports **11***a*, **12***a*, **13***a*, **14***a* of the suspension hydraulic cylinders **11**, **12**, **13**, **14** via the pipes P1, P2, P3, P4, respectively. The rolling control cylinders **31**, **32**, **33**, **34** include respective piston surfaces **31***a*, **32***a*, **33***a*, **34***a*, which have substantially the same pressure-receiving area.

The rolling control cylinders **31**, **34** are respectively connected to the suspension hydraulic cylinders **11**, **14** that are located diagonally in the vicinity of the front, left wheel and the rear, right wheel, such that the hydraulic pressures in the cylinders **31**, **34** change in the opposite way. Namely, the pistons **31***a*, **34***a* move in the opposite directions in accordance with increases and decreases in the hydraulic pressures in the cylinders **31**, **34**. The rolling control cylinders **31**, **34** constitute a right-versus-left rolling control cylinder **30**A, in which the piston surfaces **31***a*, **34***a* of the rolling control cylinders **31**, **34** are formed as an integral, common member.

On the other hand, the rolling control cylinders **32**, **33** are respectively connected to the suspension hydraulic cylinders **12**, **13** that are located diagonally in the vicinity of the front, right wheel and the rear, left wheel, such that the hydraulic pressures in the cylinders **32**, **33** change in the opposite way. Namely, the piston surfaces **32***a*, **33***a* move in the opposite directions in accordance with increases and decreases in the hydraulic pressures in the cylinders **32**, **33**. The rolling control cylinders **32**, **33** constitute a right-versus-left rolling cylinder **30**B, in the piston surfaces **32***a*, **33***a* of the rolling control cylinders **32**, **33** are formed as an integral, common number.

The right-versus-left rolling control cylinders **30**A, **30**B are arranged in the same phase and the piston surfaces **31***a*, **34***a* and the piston surfaces **32***a*, **33***a* are connected to each other by a coupling rod **35**, such that the piston surfaces **31***a*, **34***a* and the piston surfaces **32***a*, **33***a* are both pushed to the right as viewed in FIG. 1 when the hydraulic pressures of both of the left-side suspension hydraulic cylinders **11**, **13** increase, for example. The coupling rod **35** extends beyond the cylinders **31**–**34**, and its extended end portion is connected to one end of a coil spring **36** that serves as a spring element, and is also connected to one end of a shock absorber **37** that serves as a damping element for suppressing or damping vibration of the spring element. With this arrangement, the coil spring **36** and the shock absorber **37**

cooperate with each other to restrict or control the motion (i.e., axial movement) of the connecting rod 35.

The pitching controller 40 functions to control the motions of the suspension hydraulic cylinders 11, 12, 13, 14 when pitching as another type of vehicle behavior occurs, and includes pitching control cylinders 41, 42, 43, 44 that are respectively connected to the ports 11*a*, 12*a*, 13*a*, 14*a* of the suspension hydraulic cylinders 11, 12, 13, 14 via the pipes P1, P2, P3, P4, respectively. The pitching control cylinders 41, 42, 43, 44 include respective piston surfaces 41*a*, 42*a*, 43*a*, 44*a*, which have substantially the same pressure-receiving area.

The pitching control cylinders 41, 44 are respectively connected to the suspension hydraulic cylinders 11, 14 that are located diagonally in the vicinity of the front, left wheel and the rear, right wheel, such that the hydraulic pressures in the cylinders 41, 44 change in the opposite way. Namely, the piston surfaces 41*a*, 44*a* move in the opposite directions in accordance with increases and decreases in the hydraulic pressures within the cylinders 41, 44. The pitching control cylinders 41, 44 constitute a front-versus-rear pitching control cylinder 40A, in which the piston surfaces 41*a*, 44*a* of the pitching control cylinders 41, 44 are formed as an integral, common member.

On the other hand, the pitching control cylinders 42, 43 are respectively connected to the suspension hydraulic cylinders 12, 13 that are located diagonally in the vicinity of the front, right wheel and the rear, left wheel, such that the hydraulic pressures in the cylinders 42, 43 change in the opposite way. Namely, the piston surfaces 42*a*, 43*a* move in the opposite directions in accordance with increases and decreases in the hydraulic pressures within the cylinders 42, 43. The pitching control cylinders 42, 43 constitute a front-versus-rear pitching control cylinder 40B, in which the piston surfaces 42*a*, 43*a* of the pitching control cylinders 42, 43 are formed as an integral, common member.

The front-versus-rear pitching control cylinders 40A, 40B are arranged in the same phase and the piston 41*a*, 44*a* and the piston 42*a*, 43*a* are connected to each other by a coupling rod 45 such that the piston 41*a*, 44*a* and the piston 42*a*, 43*a* are both pushed to the right as viewed in FIG. 1 when the hydraulic pressures of both of the front-side suspension hydraulic cylinders 11, 12 increase, for example. The coupling rod 45 extends beyond the cylinders 41–44, and its extended end portion is connected to one end of a coil spring 46 that serves as a spring element, and is also connected to one end of a shock absorber 47 that serves as a damping element for suppressing or damping vibration of the spring element. With this arrangement, the coil spring 46 and the shock absorber 47 cooperate with each other to restrict or control the motion (i.e., axial movement) of the coupling rod 45.

In the vehicle suspension system of the embodiment constructed as described above, the suspension hydraulic cylinders 11, 12, 13, 14 make substantially the same movements (compressing motions) when the vehicle body bounces, as illustrated in FIG. 2. As a result, substantially the same hydraulic pressures (high pressures) are supplied from the ports 11*a*, 12*a*, 13*a*, 14*a* to the corresponding control cylinders 21–24, 31–34 and 41–44 through the pipes P1, P2, P3, P4, respectively.

In this condition, the hydraulic pressures in each pair of the control cylinders 31, 34, 32, 33, 41, 44 and 42, 43 of the rolling controller 30 and the pitching controller 40 are balanced with each other, and each of the pistons 31*a* (34*a*), 32*a* (33*a*), 41*a* (44*a*) and 42*a* (43*a*) makes substantially no movement. In the bouncing controller 20, on the other hand,

the pistons 21*a*, 22*a*, 23*a*, 24*a* move under the operations of the accumulator 25 and the variable restrictor 26, thereby to suppress or retard the motions of the suspension hydraulic cylinders 11, 12, 13, 14. In this manner, the bouncing controller 20 suppresses bouncing of the vehicle body, and also mitigates shocks from the road surface

When the vehicle body rolls, for example, when the vehicle turns left, the right-side suspension hydraulic cylinders 12, 14 make substantially the same movements (compressing motions) while the left-side suspension hydraulic cylinders 11, 13 make substantially the same movements (expanding motions), as illustrated in FIG. 3. As a result, substantially the same hydraulic pressures (high pressures) are supplied from the ports 12*a*, 14*a* of the right-side suspension hydraulic cylinders 12, 14 to each pair of the control cylinders 22, 24, 32, 34, and 42, 44 through the pipes P2, P4, respectively, and substantially the same hydraulic pressures (low pressures) are supplied from each pair of the control cylinders 21, 23, 31, 33, and 41, 43 to the ports 11*a*, 13*a* of the left-side suspension hydraulic cylinders 11, 13 through the pipes P1, P3, respectively.

In this condition, the hydraulic pressures in the control cylinders 21, 24 are balanced with those in the control cylinders 22, 23 in the bouncing controller 20, and the hydraulic pressures in the control cylinders 41, 44 are balanced with those in the control cylinders 42, 43 in the pitching controller 40, so that each of the pistons 21*a*, 24*a*, 22*a*, 23*a*, 41*a* (44*a*), 42*a* (43*a*) makes substantially no movement. In the rolling controller 30, on the other hand, the pistons 31*a* (34*a*) and 32*a* (33*a*) connected to each other by the coupling rod 35 move under the operations of the coil spring 36 and the shock absorber 37, thereby to suppress or retard the motions of the suspension hydraulic cylinders 11, 12, 13, 14. In this manner, the rolling controller 30 suppresses rolling of the vehicle body.

When the vehicle body undergoes pitching, for example, when the vehicle dives, the front-side suspension hydraulic cylinders 11, 12 make substantially the same movements (compressing motions) while the rear-side suspension hydraulic cylinders 13, 14 make substantially the same movements (expanding motions), as illustrated in FIG. 4. As a result, substantially the same hydraulic pressures (high pressures) are supplied from the ports 11, 12*a* of the front-side suspension hydraulic cylinders 11, 12 to each pair of the control cylinders 21, 22, 31, 32 and 41, 42 through the pipes P1, P2, respectively, and substantially the same hydraulic pressures (low pressures) are supplied from each pair of the control cylinders 23, 24, 33, 34 and 43, 44 to the ports 13*a*, 14*a* of the rear-side suspension hydraulic cylinders 13, 14 through the pipes P3, P4, respectively.

In this condition, the hydraulic pressures in the control cylinders 21, 24 are balanced with those in the control cylinders 22, 23 in the bouncing controller 20, and the hydraulic pressures in the control cylinders 31, 34 are balanced with those in the control cylinders 32, 33 in the rolling controller 30, so that each of the pistons 21*a*, 24*a*, 22*a*, 23*a*, 31*a* (34*a*), 32*a* (33*a*) makes substantially no movement. In the pitching controller 40, on the other hand, the pistons 41*a* (44*a*) and 42*a* (43*a*) connected to each other by the coupling rod 45 move under the operations of the coil spring 46 and the shock absorber 47, thereby to suppress or retard the motions of the suspension hydraulic cylinders 11, 12, 13, 14. In this manner, the pitching controller 40 suppresses pitching of the vehicle body.

When the vehicle twists on an unleveled ground, for example, the front-right and rear-left suspension hydraulic cylinders 12, 13 make substantially the same movements

(compressing motions) while the front-left and rear-right suspension hydraulic cylinders **11**, **14** make substantially the same movements (expanding motions), as illustrated in FIG. 5. As a result, substantially the same hydraulic pressures (neutral hydraulic pressures as shown in FIG. 1) are supplied from the ports **12***a*, **13***a* of the suspension hydraulic cylinders **12**, **13** to each pair of the control cylinders **22**, **23**, **32**, **33** and **42**, **43** through the pipes P2, P3, respectively, and substantially the same hydraulic pressures (neutral hydraulic pressures as shown in FIG. 1) are supplied from each pair of the control cylinders **21**, **24**, **31**, **34** and **41**, **44** to the ports **11***a*, **14***a* of the suspension hydraulic cylinders **11**, **14** through the pipes P1, P4, respectively.

In this condition, the hydraulic pressures in the control cylinders **31**, **34**, **32**, **33** are balanced with each other in the rolling controller **30**, and the hydraulic pressure in the control cylinders **41**, **44**, **42**, **43** are balanced with each other in the pitching controller **40**, so that each of the pistons **31***a* (**34***a*), **32***a* (**33***a*), **41***a* (**44***a*), **42***a* (**43***a*) makes substantially no movement. In the bouncing controller **20**, on the other hand, hydraulic fluid is supplied to each of the control cylinders **22**, **23**, and hydraulic fluid is discharged from each of the control cylinders **21**, **24** whereby the pistons **21***a*, **24***a* move in the same direction as the pistons **22***a*, **23***a*. However, since the moving amount of the pistons **21***a*, **24***a* is the same as that of the pistons **22***a*, **23***a*, the bouncing controller **20** does not substantially function. Namely, the bouncing controller **20** does not operate to suppress the motions of the suspension hydraulic cylinders **11**, **12**, **13**, **14**.

As is apparent from the above description, the vehicle suspension system of this embodiment is constructed such that the motions of the suspension hydraulic cylinders **11**, **12**, **13**, **14** are independently restricted or controlled by the bouncing controller **20** including the accumulator **25** (spring element) and the variable restrictor **26** (damping element), the rolling controller **30** including the coil spring **36** (spring element) and the shock absorber **37** (damping element), and the pitching controller **40** including the coil spring **46** (spring element) and the shock absorber **47** (damping element). It is also possible to independently set the characteristics of the respective spring elements and damping elements that specify the behavior controlling (or restricting) functions of the controllers **20**, **30**, **40**. Thus, the characteristics of the spring and damping elements of each controller **20**, **30**, **40** can be independently set to those suitable for each of plural types of behavior of the vehicle body, and therefore each type of behavior can be optimally suppressed.

In the vehicle suspension system of this embodiment, a hydraulic circuit is constructed by simply connecting the single ports **11***a*–**14***a* of the suspension hydraulic cylinders **11**–**14** mounted for the front and rear, right and left wheels, to the corresponding cylinders **21**–**24**, **31**–**34** and **41**–**44**, via the respective pipes P1–P4. Thus, the hydraulic circuit can be made simple and inexpensive. Furthermore, the vehicle suspension system of the embodiment is able to not only effectively suppress the behavior (i.e., bouncing) of the vehicle body in the heaving direction, but also suitably deal with the situation where a force that twists the vehicle body is applied to the front and rear, right and left wheels when the vehicle is traveling on an unleveled ground, for example. More specifically, when the vehicle body twists on such an unleveled ground, the diagonal hydraulic control cylinders **20**A, **20**B freely operate in the same phase (namely, the pistons **21***a*, **24***a*, **22***a*, **23***a* move in the same direction) without actuating the accumulator **25** provided in the bouncing controller **20**, and therefore the vertical load and driving force measured at each wheel are less likely to be reduced.

Thus, the vehicle suspension system permits the vehicle posture or attitude to be favorably maintained while assuring sufficient driving force of each wheel, without making the hydraulic circuit undesirably complicated.

In the bouncing controller **20** of the vehicle suspension system of this embodiment, the pistons **21***a*, **24***a* of the bouncing control cylinders **21**, **24** that constitutes the diagonal hydraulic control cylinder **20**A are coupled to each other, and the pistons **22***a*, **23***a* of the bouncing control cylinders **22**, **23** that constitute the diagonal hydraulic control cylinder **20**B are coupled to each other. Thus, the diagonal hydraulic control cylinders **20**A, **20**B can be made compact or small-sized.

In the vehicle suspension system of this embodiment, the coupling device **20**C that couples the diagonal hydraulic control cylinders **20**A, **20**B in the bouncing controller **20** includes the accumulator **25** and the variable restrictor **26**, and takes the form of a liquid-tight coupling structure using a hydraulic fluid as a medium. The coupling device **20**C may be modified such that a hydraulic fluid oil is also supplied to or drained from the hydraulic chamber **27** that communicates with the accumulator **25** via the variable restrictor **26** (or the hydraulic chamber **25***a* of the accumulator **25**) in accordance with, for example, the load of the vehicle body, so that the vehicle height can be adjusted while maintaining the vehicle posture or attitude.

In the vehicle suspension system of this embodiment, the rolling controller **30** and the pitching controller **40** are provided in addition to the bouncing controller **20**. This arrangement makes it possible to suppress or control the behavior (rolling) of the vehicle body in the rolling direction and the behavior (pitching) of the vehicle body in the pitching direction, as well as the behavior (bouncing) of the vehicle body in the heaving direction.

In the vehicle suspension system of this embodiment, at least one of the variable restrictor **26** of the bouncing controller **20**, a variable restrictor **37***a* included in the shock absorber **37** of the rolling controller **30**, and a variable restrictor **47***a* included in the shock absorber **47** of the pitching controller **40** may be provided with a characteristic switching mechanism (i.e., actuator). With the characteristic switching mechanism thus provided, semi-active control of the damping force can be performed with respect to each type of behavior of the vehicle body, thus assuring further improved vehicle riding comfort. In this case, the maximum number of required actuators is three, which is less by one than four actuators that would be otherwise required in the case where semi-active control of the damping force is independently performed with respect to the four wheels.

In the vehicle suspension system of this embodiment, an actuator may be provided for controlling increases and decreases in the hydraulic pressure of the hydraulic chamber **27** in the bouncing controller **20**, or an actuator may be provided for controlling increases and decreases in the spring force of the coil spring **36** in the rolling controller **30**, or an actuator may be provided for controlling increases and decreases in the spring force of the coil spring **46** in the pitching controller **40**. With such actuator or actuators provided, the vehicle posture or attitude can be suitably controlled.

In this case, if the bouncing controller **20** is semi-actively operated, and the rolling controller **30** and the pitching controller **40** are actively operated, for example, the semi-actively operating bouncing controller **20** is able to support the load of the vehicle body, thus eliminating a need for the actively operating rolling controller **30** and pitching controller **40** to support the vehicle body load. This leads to a

reduction in the size of actuators (or other power sources) employed for actively operating the rolling controller **30** and the pitching controller **40**, resulting in a reduction of the energy consumed by the actuators.

In the illustrated embodiment, the coupling device **20C** for coupling the diagonal hydraulic control cylinders **20A**, **20B** in the bouncing controller **20** includes the accumulator **25**, the variable restrictor **26**, and the hydraulic cylinder **27**. However, the invention is not limited to this arrangement of the coupling device **20C**. As shown in FIG. **6**, for example, the coupling device **20C** for coupling the diagonal hydraulic control cylinders **20A**, **20B** in the bouncing controller **20** may include a coil spring **28** and a shock absorber **29**. In the modified embodiment of FIG. **6**, too, the bouncing controller **20**, the rolling controller **30** and the pitching controller **40** may be semi-actively or actively operated as needed, in a manner similar to that of the illustrated embodiment.

In the illustrated embodiment, the pipes **P1**, **P2**, **P3** and **P4** are connected in the manner as shown in FIG. **1**, to provide the effects as described above. However, the invention is not limited to this arrangement. For example, the pipes **P1**, **P2**, **P3** and **P4** may be connected as shown in FIG. **7** so as to provide effects similar to those of the illustrated embodiment. It is to be noted that the embodiment as shown in FIG. **7** is identical with that of FIG. **1** except the constructions of the right-versus-left rolling control cylinders **30A**, **30B** of the rolling controller **30** and the front-versus-rear pitching control cylinders **40A**, **40B** of the pitching controller **40**.

In the right-versus-left rolling control cylinder **30A** as shown in FIG. **7**, the rolling control cylinders **31**, **32** are connected to each other such that the hydraulic pressures in the rolling control cylinders **31**, **32** change in the opposite way, and the pistons **31a**, **32a** of the rolling control cylinders **31**, **32** are integrated into a single, common piston. In the right-versus-left rolling control cylinder **30B**, the rolling control cylinders **33**, **34** are connected to each other such that the hydraulic pressures in the rolling control cylinders **33**, **34** change in the opposite way, and the pistons **33a**, **34a** of the rolling control cylinders **33**, **34** are integrated into a single, common piston.

In the front-versus-rear pitching control cylinder **40A** as shown in FIG. **7**, on the other hand, the pitching control cylinders **42**, **44** are connected to each other such that the hydraulic pressures in the pitching control cylinders **42**, **44** change in the opposite way, and the pistons **42a**, **44a** of the pitching control cylinders **42**, **44** are integrated into a single, common piston. In the front-versus-rear pitching control cylinder **40B**, the pitching control cylinders **41**, **43** are connected to each other such that the hydraulic pressures in the pitching control cylinders **41**, **43** change in the opposite way, and the pistons **41a**, **43a** of the pitching control cylinders **41**, **43** are integrated into a single, common piston.

In the illustrated embodiments, it is assumed that substantially the same hydraulic pressure is supplied to appropriate ones of the control cylinders (**21–24**, **31–34**, **41–44**) activated upon bouncing, rolling or pitching of the vehicle body. On the basis of this assumption, each of the pistons (**21a–24a**, **31a–34a**, **41a–44a**) is provided with substantially the same pressure-receiving area, so as to achieve a desired balance in the hydraulic pressure. If the vehicle suspension system is constructed such that substantially the same hydraulic pressure is not supplied to appropriate ones of the control cylinders at the time of bouncing, rolling or pitching, the pressure-receiving area of each piston may be set to a suitable value, so as to achieve a desired balance in the hydraulic pressure. Thus, the pressure-receiving area of each of the pistons (**21a–24a**, **31a–34a**, **41a**, **44a**) need not be set to substantially the same value.

In the illustrated embodiment, the vehicle suspension system includes three behavior controllers, i.e., the bouncing controller **20**, the rolling controller **30**, and the pitching controller **40**. However, the vehicle suspension system may include only two behavior controllers, e.g., the bouncing controller **20** and the rolling controller **30**, or the bouncing controller **20** and the pitching controller **40**, or the rolling controller **30** and the pitching controller **40**.

What is claimed is:

1. A vehicle suspension system of a motor vehicle, comprising:
   - a plurality of suspension devices mounted on the vehicle with respect to right and left wheels of the vehicle, respectively;
   - a rolling controller that controls a motion of each of the suspension devices when a vehicle body undergoes a rolling;
   - a pitching controller that controls a motion of each of the suspension devices when the vehicle body undergoes a pitching, independently of the rolling controller; and
   - a bouncing controller that controls a motion of each of the suspension devices during bouncing of the vehicle body,

   wherein the bouncing controller, rolling controller, and pitching controller each operate independently and each controller includes an accumulator or a spring as a spring element and a variable resistor or a shock absorber as a damping element, the spring element and the damping element defining a behavior suppression function for each controller.

2. A vehicle suspension system of a motor vehicle, comprising:
   - a plurality of suspension hydraulic cylinder portions mounted on the vehicle with respect to front-right, front-left, rear-right, and rear-left wheels of the vehicle, each of the suspension hydraulic cylinder portions having a single port; and
   - a plurality of bounce control hydraulic cylinder portions each of which is connected to the single port of a corresponding one of the suspension hydraulic cylinder portions via a pipe, for controlling a motion within the corresponding suspension hydraulic cylinder portion, wherein the plurality of the bounce control hydraulic cylinder portions are connected to form two pairs of diagonal hydraulic control cylinder portions such that hydraulic pressures in two of the bounce control hydraulic cylinder portions connected to diagonally located ones of the suspension hydraulic cylinder portions change in substantially the same way, the two pairs of diagonal hydraulic control cylinder portions being opposed to each other and connected to a device capable of controlling motions within the diagonal hydraulic control cylinder portions,

   the system further comprising:
   - a rolling controller that controls rolling of a body of the vehicle; and
   - a pitching controller that controls pitching of the vehicle body,

   wherein the rolling controller and the pitching controller operate independently from each other.

3. The vehicle suspension system according to claim **2**, wherein the plurality of bounce control hydraulic cylinder portions comprises a first pair of bounce control hydraulic cylinder portions connected to the suspension hydraulic cylinders for front-left and rear-right wheels, and a second

pair of bounce control hydraulic cylinder portions connected to the suspension hydraulic cylinder portions for front-right and rear-left wheels, and wherein the first pair of bounce control hydraulic cylinder portions form a first unit and the second pair of bounce control hydraulic cylinder portions form a second unit to provide the two pairs of the diagonal hydraulic control cylinder portions.

4. The vehicle suspension system according to claim 2, wherein pistons received in each pair of the control hydraulic cylinder portions are coupled to each other such that the pistons are movable as a unit.

5. The vehicle suspension system according to claim 2, including a motion controlling device comprising a liquid-tight structure including an accumulator and using a hydraulic fluid as a medium.

6. The vehicle suspension system according to claim 2, including a motion controlling device comprising a coil spring and a shock absorber which are located between the two pairs of diagonal hydraulic control cylinder portions.

7. The vehicle suspension system according to claim 2, wherein pistons received in the control hydraulic cylinder portions have substantially the same pressure-receiving area.

8. The vehicle suspension system according to claim 2, the rolling controller comprising a plurality of rolling control cylinder portions each of which is connected to the single port of a corresponding one of the suspension hydraulic cylinder portions via a pipe, for controlling a motion within the corresponding suspension hydraulic cylinder portion,

wherein the rolling control cylinder portions form a pair of rolling control cylinders such that hydraulic pressures in two of the plurality of rolling control cylinder portions connected to diagonally located ones of the suspension hydraulic cylinder portions change in opposite directions, the pair of rolling control cylinders being coupled by a coupling rod such that the plurality of rolling control cylinder portions associated with the front-right and rear-right wheels are oriented in the same direction and the plurality of rolling control cylinder portions associated with the front-left and rear-left wheels are oriented in the same direction, the coupling rod being coupled to a motion control device that controls a motion of the coupling rod.

9. The vehicle suspension system according to claim 8, wherein the motion control device comprises a spring element and a damping element that are connected to one end of the coupling rod.

10. The vehicle suspension system according to claim 9, wherein the motion control device comprises a coil spring serving as the spring element and a shock absorber serving as the damping element.

11. The vehicle suspension system according to claim 8, wherein the two pairs of diagonal hydraulic control cylinder portions and the pair of rolling control cylinders operate independently of each other.

12. The vehicle suspension system according to claim 2, the pitching controller comprising a plurality of pitching control cylinder portions each of which is connected to the single port of a corresponding one of the suspension hydraulic cylinder portions via a pipe, for controlling a motion within the corresponding suspension hydraulic cylinder portion,

wherein the pitching control cylinder portions form a pair of pitching control cylinders such that hydraulic pressures in two of the plurality of pitching control cylinder portions connected to diagonally located ones of the suspension hydraulic cylinder portions change in opposite directions, the pair of pitching control cylinders being coupled by a coupling rod such that the plurality of pitching control cylinder portions associated with the front-right and rear-right wheels are oriented in opposite directions and the plurality of pitching control cylinder portions associated with the front-left and rear-left wheels are oriented in opposite directions, the coupling rod being coupled to a motion control device that controls a motion of the coupling rod.

13. The vehicle suspension system according to claim 12, wherein the motion control device comprises a spring element and a damping element that are connected to one end of the coupling rod.

14. The vehicle suspension system according to claim 13, wherein the motion control device comprises a coil spring serving as the spring element and a shock absorber serving as the damping element.

15. The vehicle suspension system according to claim 12, wherein the two pairs of diagonal hydraulic control cylinder portions and the pair of pitching control cylinders operate independently of each other.

16. The vehicle suspension system according to claim 2, wherein:

the rolling controller comprising a plurality of rolling control cylinder portions each of which is connected to the single port of a corresponding one of the suspension hydraulic cylinder portions via a pipe, for controlling a motion within the corresponding suspension hydraulic cylinder portion;

the pitching controller comprising a plurality of pitching control cylinder portions each of which is connected to the single port of a corresponding one of the suspension hydraulic cylinder portions via a pipe, for controlling a motion within the corresponding suspension hydraulic cylinder portion,

wherein the rolling control cylinder portions form a pair of rolling control cylinders such that hydraulic pressures in two of the plurality of rolling control cylinder portions connected to diagonally located ones of the suspension hydraulic cylinder portions change in opposite directions, the pair of rolling control cylinders being coupled by a first coupling rod such that the plurality of rolling control cylinder portions associated with the front-right and rear-right wheels are oriented in the same direction and the plurality of rolling control cylinder portions associated with the front-left and rear-left wheels are oriented in the same direction, the first coupling rod being coupled to a first motion control device that controls a motion of the coupling rod; and

wherein the pitching control cylinder portions form a pair of pitching control cylinders such that hydraulic pressures in two of the plurality of pitching control cylinder portions connected to diagonally located ones of the suspension hydraulic cylinder portions change in opposite directions, the pair of pitching control cylinders being coupled by a second coupling rod such that a plurality of the pitching control cylinder portions associated with the front-right and rear-right wheels are oriented in opposite directions and the plurality of pitching control cylinder portions associated with the front-left and rear-left wheels are oriented in opposite directions, the second coupling rod being coupled to a second motion control device that controls a motion of the second coupling rod.

17. The vehicle suspension system according to claim 16, wherein the first motion control device includes a spring

element and a damping element that are connected to one end of the first coupling rod, and the second motion control device includes a spring element and a damping element that are connected to one end of the second coupling rod.

18. The vehicle suspension system according to claim **17**, wherein each of the first and second motion control devices comprises a coil spring serving as the spring element and a shock absorber serving as the damping element.

19. The vehicle suspension system according to claim **16**, wherein the two pairs of diagonal hydraulic control cylinder portions, the pair of rolling control cylinders and the pair of pitching control cylinders operate independently of one another.

20. The vehicle suspension system according to claim **2**, wherein the bounce control hydraulic cylinder portions operate independently from the rolling controller and the pitching controller.

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